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ANNEX

Holly

3-VIII

U. S. E. DUPLICATE
191

SECOND EDITION.

OFFICIAL REPORTS

OF

VARIOUS DUTY TRIALS

OF THE

Gaskill Pumping Engine

THE HOLLY MANUFACTURING CO.

LOCKPORT, N. Y.

Pumping Eng

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TILDEN FOUNDATIONS
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THE HOLLY MANUFACTURING COMPANY has been in the manufacturing business since 1859, and since 1863 in making pumping machinery and building water works. It has been foremost in adopting and bringing out valuable improvements, and on three occasions made such departures from old lines in pumping engines as involved a complete abandonment of plans and patterns which had cost tens of thousands of dollars, to take up a new and better machine. This spirit of enterprise has been well rewarded with success. From a small beginning, with a cash capital of $20,000 and a few hands, it has grown to have $800,000 capital and surplus, and more than five hundred men on its weekly pay-roll, all exclusively devoted to pumping machinery for water works.

The latest departure was made in 1882, when the then well-known and highly-approved Holly Quadruplex Compound Condensing Pumping Engine was abandoned for the present famous Gaskill Pumping Engine.

The incompressibility of water renders it necessary, in order to secure the best results in operating reciprocating pumps, to run them at a comparatively low rate of speed, while the expansive quality of steam admits of a high velocity, which is the most economical in reciprocating engines. In the application of steam to pumping water, the question, how to harmonize these opposite conditions, has been the subject of much study, many theories and numerous practical experiments, until it is now almost universally admitted that the compound steam engine is the most satisfactory method.

Several types and various designs of compound pumping engines have been produced during the past thirty years by scientific and practical engineers, some of which have peculiar merits of greater or less utility and others important features of undoubted practical value. A few were complicated in their mechanism, elaborate in design, costly in construction, expensive in maintenance, but showing high economy in the use of steam. Others were simple and cheap, but not so economical in the use of steam, which it was claimed would, considering first cost, interest on the money invested and expense of repairs, be the cheapest in the end.

In designing the pumping engine herein described, Mr. Gaskill had in view the production of one which would combine simplicity of construction, low first cost, with high economy in steam. The following pages show conclusively that he met with signal success.

The Gaskill Horizontal Pumping Engine was first introduced at the Saratoga Springs, New York, Water Works, by The Holly Manufacturing Company under a contract with the Water Commissioners which specified that "Said pumping engine "shall have a pumping capacity to deliver four million United States gallons of water "in twenty-four hours against eighty pounds domestic pressure to the square inch at "eighteen revolutions per minute, and guarantee that said engine can be run at thirty "revolutions per minute with perfect safety; said pumping engine shall with safety to "all its parts carry a fire pressure of one hundred and forty pounds to the square inch, "and develop a duty equal to raising eighty million pounds of water one foot high "with one hundred pounds of best coal consumed, and an average daily duty of sixty-"five million foot-pounds with good merchantable anthracite coal, reasonably free from "dirt and slate."

The results of the two separate trials, one conducted by J. W. Hill, M. E., and Prof. D. M. Greene; the other by Prof. Chas. T. Porter; and the daily record of operating engineers David L. Holland and George F. Bacon, all of which are published in another part of this book, show that the guarantees were not only fulfilled but largely exceeded.

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DESCRIPTION OF THE GASKILL PUMPING ENGINE.

On a heavy iron bed plate are mounted two pumps, and in direct line therewith two low-pressure steam cylinders (see following plates, figs. 1, 2 and 3), with the piston rods of the low-pressure steam cylinders connected to the piston rods of the pumps. Between the pumps and steam cylinders are placed two beam supports (figs. 4, 5 and 6), which are firmly bolted to the bed plate and rigidly stayed by wrought-iron struts to the pumps and steam cylinders. These beam supports carry the beam shafts and beams (figs. 7, 8 and 9), the lower end of the latter being connected to the cross heads of the low-pressure cylinders by means of links. On top of the pumps are placed the main shaft bearings, which support the main shaft, a fly-wheel and two cranks, the latter being keyed to the shaft at right angles to each other. On top of the low-pressure steam cylinders are mounted two high-pressure steam cylinders, with their centers in the same horizontal plane as the center of the main crank shaft. Two cross heads (figs. 10, 11, 12 and 13), for the high-pressure steam cylinders are connected by means of links to the upper ends of the beams, and the beams are in turn connected by means of connecting rods to two crank pins. From the high-pressure steam cylinders heavy cast-iron girders extend to the main shaft bearings or pillow blocks. On the inner end of each of the beam centers an arm is keyed from which the air pumps are driven. The valves of the steam cylinders are operated by means of eccentrics (figs. 18 and 19) keyed on two shafts, which are at right angles with and driven by the main shaft through small bevel gears. The admission valves to the high-pressure steam cylinders are of the double beat puppet pattern, so arranged (figs. 18 and 19) as to open at the proper time and to close at any desired point of the stroke. The exhaust valves from the high-pressure cylinders serve also as admission valves to the low-pressure cylinders, are of the ordinary slide-valve type, and so set as to remain open somewhat less time than is required to make a complete stroke. The exhaust valves from the low-pressure cylinders are also plain slide valves, operating in the same manner as the high-pressure exhaust valves.

The plungers are arranged to work through glands in the center of the pumps, and are accessible from the covers at the ends of the pump cylinders. The pump valves are placed on horizontal plates below and above the line of plunger travel. The glands divide the valves of one end of the pump from those of the other end, at the center of the valve plates.

The operation of the machine is as follows:

Let it be understood (1) that this arrangement constitutes two distinct pumping engines coupled together through the crank shaft; (2) that the two engines are usually worked together, but (3) that either engine may be uncoupled and operated independently of the other.

Steam is admitted through the automatic cut-off valves into the high-pressure steam cylinders, urging the pistons forward under full boiler pressure until the point of cut-off is reached. The admission valves then close and the remaining portion of the stroke is accomplished by the expansive force of the steam. When the pistons have nearly reached the end of their stroke, the exhaust valves between the high and low-pressure cylinders open and the steam remaining in the high-pressure cylinders passes immediately into the low-pressure cylinders and against their pistons, which are then at the end of their stroke and opposite the high-pressure pistons. The low-pressure pistons are then in turn urged forward by the incoming steam, which is expanded to four times the volume it had in the high-pressure cylinders at the time of its release therefrom. The release of steam from the low-pressure cylinders is accomplished in the return stroke by means of the exhaust valves. This operation is repeated by each engine and at each end alternately. The close connection between each pair of high and low-pressure cylinders reduces the clearance spaces to a minimum, which with thorough jacketing insures the most economical use of steam.

This engine is also built to operate as a non-compound engine, in which case the upper or high pressure cylinders and connections are omitted, and the lower steam cylinders provided with automatic cut-off valves. Steam is admitted to these cylinders direct from the boiler and exhausted into the condenser. This mode of construction reduces the cost and may be preferred by cities and villages where cheapness of fuel renders the price of the machine a matter to be considered rather than the annual saving of steam—although when constructed in this way a duty of 70,000,000 foot-pounds of work can be obtained from 100 pounds of coal.

The more valuable features and peculiar merits of the Gaskill Pumping Engine are as follows:

First. HIGH FUEL ECONOMY—This is shown in the following reports of duty trials and the daily record at the Saratoga pumping station, published in another part of this book.

Second. MODERATE FIRST COST—The first cost is somewhat higher than that of direct-acting pumps, but this difference is many times counterbalanced by the efficiency of the service performed, by decreased coal consumption, immunity from accidents and expense of repairs.

Third. LOW PISTON SPEED—The high duties heretofore obtained by other engines have been largely due to high speeds and high steam pressures; the results reached by the Gaskill Engine have been obtained at moderate speeds and low pressures of steam.

Fourth. SIMPLICITY OF DESIGN—An engine combining the features of compounding, automatic cut-off, crank and fly-wheel and approaching this in simplicity has never before been designed.

Fifth. ACCESSIBILITY OF PARTS—The entire machine, excepting air pumps and condensers, is placed on the line of the engine-room floor so that all parts are easy of access—especially the pumps which require no scaffolding and are provided with hand and man-holes for convenience in getting at the interior.

Sixth. PERFECT STEAM DISTRIBUTION—The high and low-pressure cylinders have short connecting ports and all admission and exhaust valves are placed close to the cylinders so that the clearance or waste spaces are reduced to the minimum. This feature contributes in a large measure to the high fuel economy obtained.

Seventh. PERFECT PUMP ACTION—The pump valves are small, have a lift of only five-sixteenths of an inch, and are noiseless in their action (see fig. 20). The slight lift prevents lost action on the reversal of the pump plunger. The number of valves in each pump is proportioned to the quantity of water designed to pass through them so that the loss by friction is inappreciable. The pumps are so arranged as to avoid abrupt changes of direction of the water; and all passages are of ample size.

Eighth. UNIFORM LENGTH OF STROKE—The motion of the pistons and plungers being controlled by a crank and fly-wheel the strokes are always of the same length, and no loss of action in the pump or waste of steam in the steam cylinders can occur from shortage of stroke as is the case in all direct-acting pumping engines.

Ninth. THOROUGH STEAM JACKETING—The steam cylinders are thoroughly steam jacketed both on heads and sides, which is essential to economy of steam.

The correctness of the foregoing statements are fully verified by practical experience, with the following list of engines in use and the accompanying reports.

Since this engine was designed, one hundred and forty-one have been sold, ranging in capacity from one million to twenty million gallons, with an aggregate capacity of over thirteen hundred million gallons (against one hundred feet head) daily, and the Company has been compelled to largely increase its facilities for their manufacture.

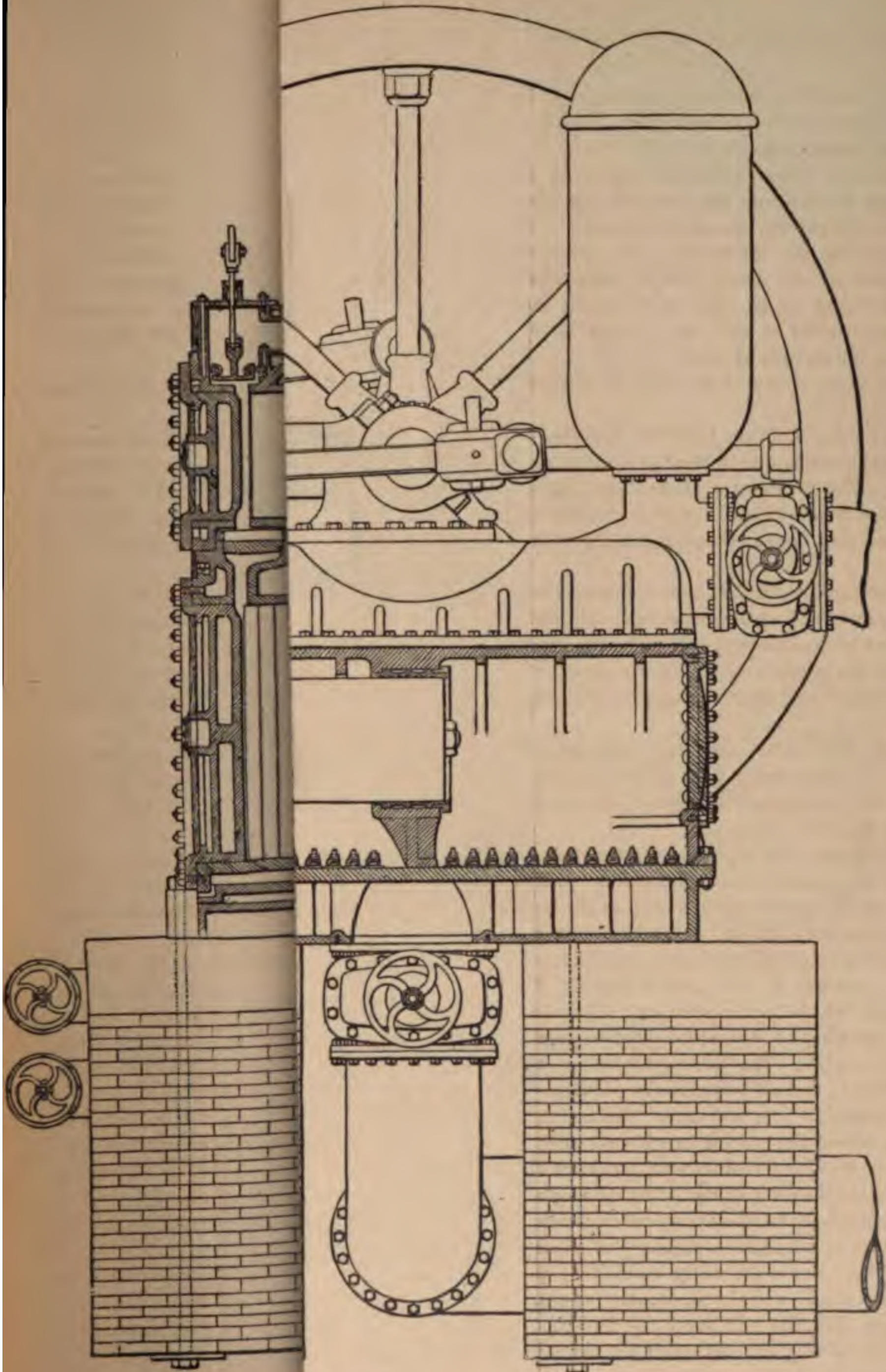

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Fig. 2.

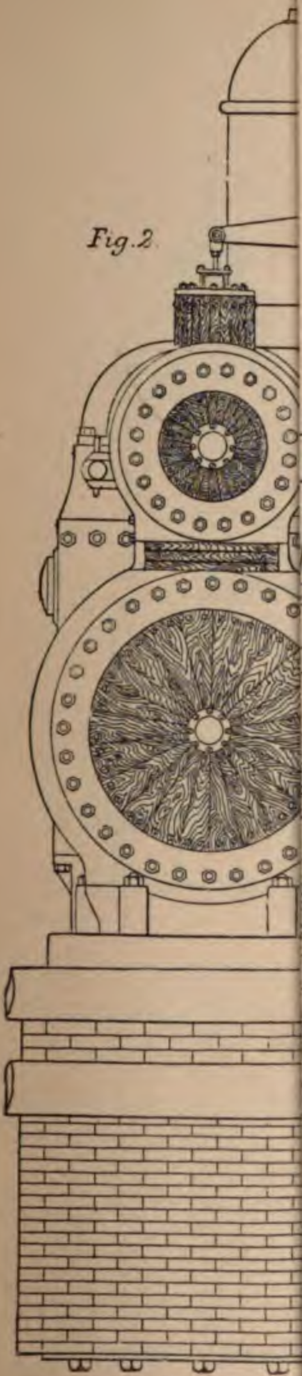
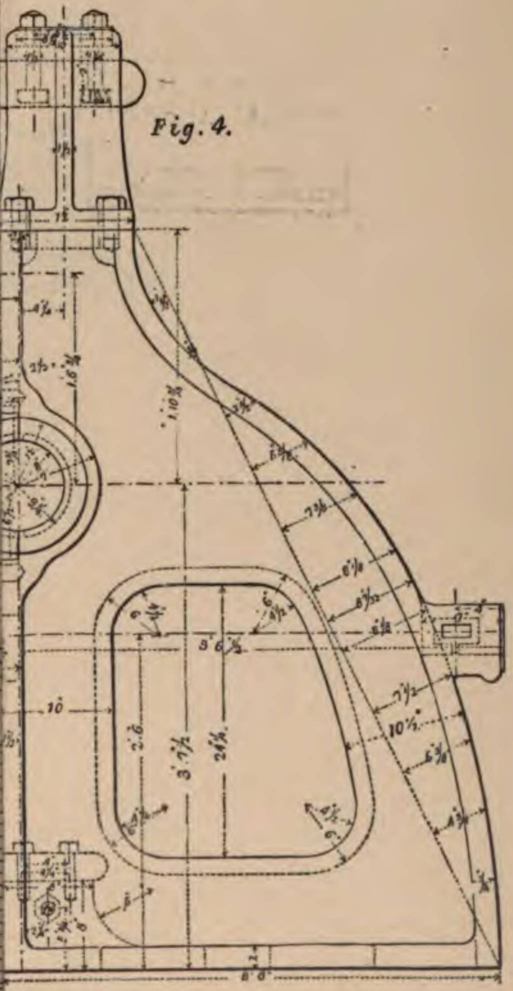


Fig. 4.



OUTSIDE. E ELEVATION OF BEAM SUPPORT.

STEAM END E

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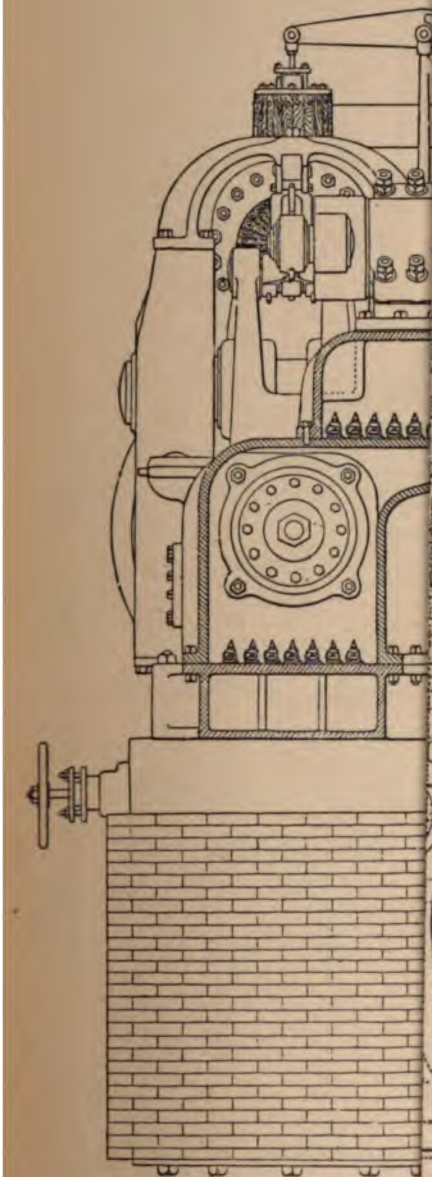
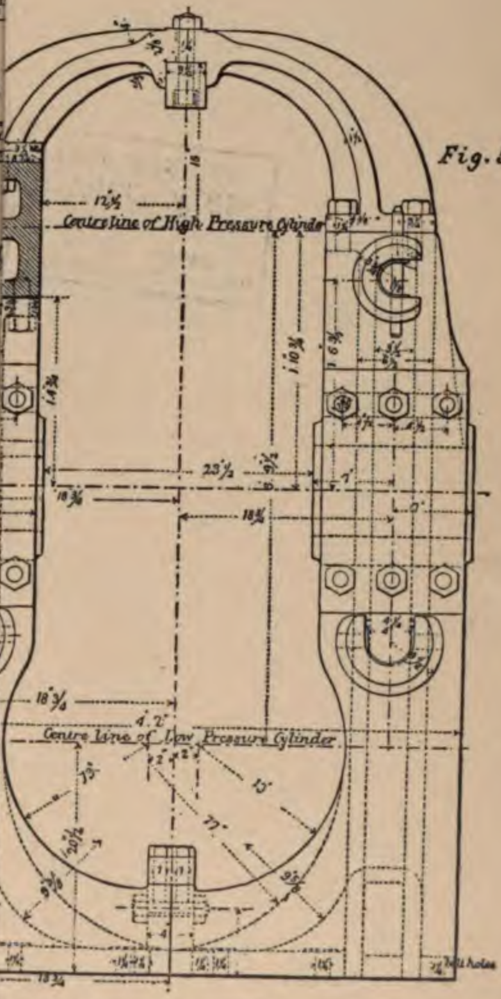
Fig. 3.



SECTIONAL.

PUMP END EL

Fig. 5.


12¼
Centre line of High Pressure Cylinder
18
1.4⅜
1.10⅜
1.6½
23½
18⅜
18½
7
18¾
4.2
Centre line of Low Pressure Cylinder
15
77
20½
9⅜
18¾
bolt holes

ELEVATION OF BEAM SUPPORT.

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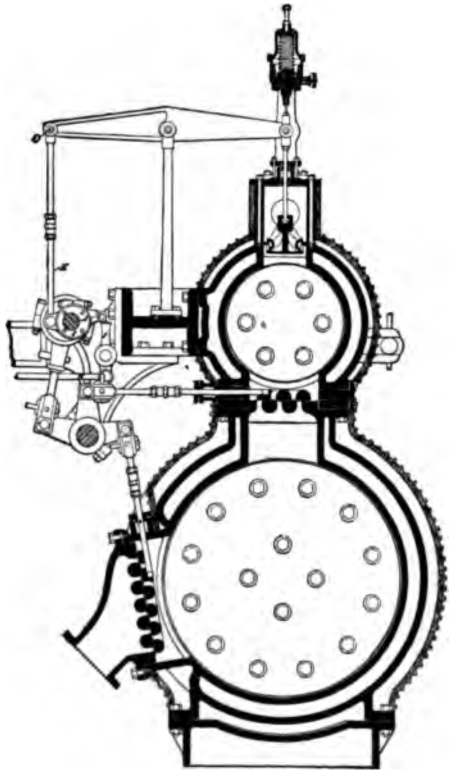
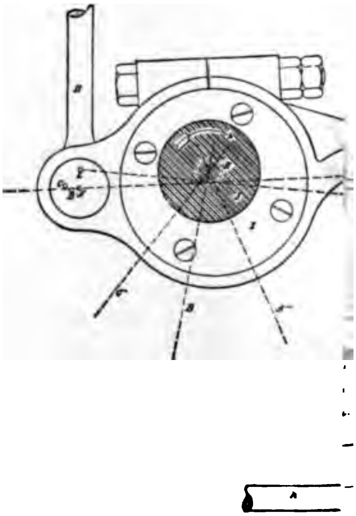


SECTIONAL ELEVATION OF STEAM CYLINDERS SHOWING VALVE MOTION.

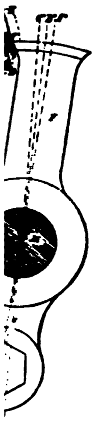
Fig. 18.



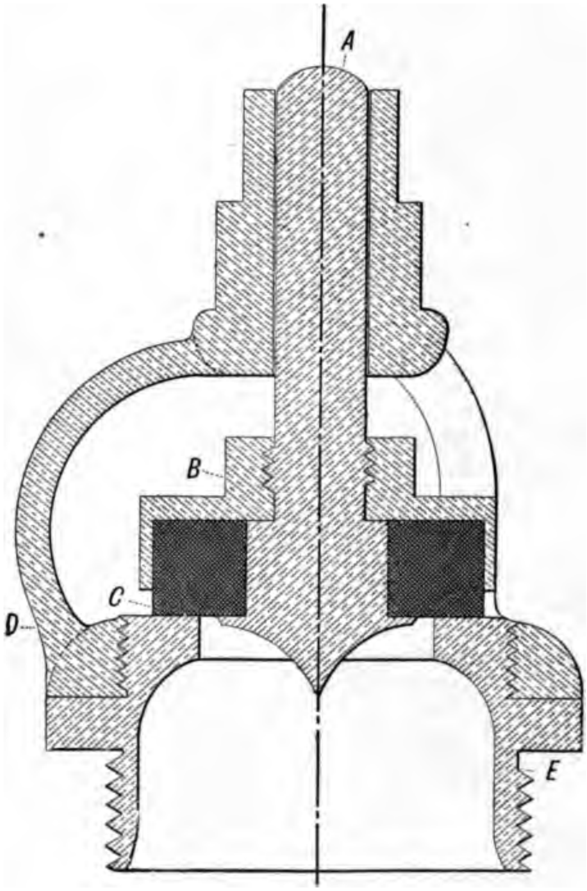
CUT-OFF ECCENTRIC AND

Fig. 19.

GASKILL PUM



AM.



SECTIONAL ELEVATION OF "TROY" PUMP VALVE.
FULL SIZE.
Fig. 20.

ENGINE

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LIST OF ENGINES BUILT BY THE HOLLY MANUFACTURING COMPANY, OF THE GASKILL CRANK AND FLY-WHEEL TYPE, SINCE 1882.

Place.	Capacity.
1. Saratoga Springs, N. Y	5,000,000
2. Des Moines, Ia	5,000,000
3. East Orange, N. J	1,500,000
4. Fredericton, N. B	1,500,000
5. Adrian, Mich	1,500,000
6. Adrian, Mich	1,500,000
7. Auburn, N. Y	7,500,000
8. Brockville, Ont	1,500,000
9. Columbus, Ohio	10,000,000
10. Doylestown, Pa	1,500,000
11. Flint, Mich	1,500,000
12. Flint, Mich	1,500,000
13. Omaha, Neb	5,000,000
14. Salina, Kan	1,500,000
15. West Bay City, Mich	1,500,000
16. West Bay City, Mich	1,500,000
17. Ashland, Wis	1,500,000
18. Burlington, N. J	1,500,000
19. Gainesville, Texas	1,500,000
20. Kansas City, Mo	4,000,000
21. Lexington, Ky	1,500,000
22. Lexington, Ky	1,500,000
23. Oshkosh, Wis	4,000,000
24. Oshkosh, Wis	4,000,000
25. Rockford, Ill	3,000,000
26. Saginaw City, Mich	4,000,000
27. Sioux City, Ia	3,000,000
28. Sioux City, Ia	3,000,000
29. Ashland, Wis	1,500,000
30. Austin, Tex	5,000,000
31. Albuquerque, N. M	3,000,000
32. Brockville, Ont	1,500,000
33. Buffalo, N. Y	15,000,000
34. Burlington, Ia	3,000,000
35. Batavia, N. Y	1,500,000
36. Fond du Lac, Wis	3,000,000
37. Fond du Lac, Wis	3,000,000
38. Jackson, Mich	4,000,000
39. Leavenworth, Kas	5,000,000
40. Norfolk, Va	2,000,000
41. Norfolk, Va	5,000,000
42. Richmond, Ind	4,000,000
43. Tiffin, Ohio	3,000,000
44. Wausau, Wis	3,000,000
45. Wausau, Wis	3,000,000
46. Chicago, Ill	12,000,000
47. Chicago, Ill	12,000,000
48. Valparaiso, Ind	1,500,000
49. Hyde Park, Ill	12,000,000
50. Tonawanda, N. Y	2,000,000
51. Tonawanda, N. Y	2,000,000
52. San Diego, Cal	2,000,000
53. Beatrice, Neb	1,500,000
54. Beatrice, Neb	1,500,000
55. Streator, Ill	1,500,000
56. Streator, Ill	1,500,000
57. Denison, Texas	1,500,000
58. Denison, Texas	1,500,000
59. Bay City, Mich	5,000,000
60. Beverly, Mass	2,000,000
61. Beverly, Mass	2,000,000
62. Lima, Ohio	3,000,000
63. Lima, Ohio	3,000,000
64. Erie, Pa	5,000,000
65. Schenectady, N. Y	3,000,000
66. Boston, Mass	8,000,000
67. Boston, Mass	8,000,000
68. Philadelphia, Pa	20,000,000
69. Joliet, Ill	3,000,000
70. Topeka, Kan	4,000,000
71. Mobile, Ala	6,000,000
72. Mobile, Ala	6,000,000
73. City of Lake View, Ill	12,000,000
74. Buffalo, N. Y	20,000,000
75. Chester, Pa	4,000,000
76. Chester, Pa	4,000,000
77. Omaha, Neb	14,000,000
78. San Diego, Cal	3,000,000
79. Evanston, Ill	5,000,000
80. Town of Lake, Ill	12,000,000
81. Hutchinson, Kan	1,500,000
82. Washington, D. C	2,500,000
83. Washington, D. C	2,500,000
84. Port Huron, Mich	5,000,000
85. Kansas City, Mo	9,000,000
86. Dallas, Texas	8,000,000
87. La Salle, Ill	2,000,000
88. Portland, Ore	1,500,000
89. Buffalo, N. Y	20,000,000
90. Columbus, Ind	3,000,000
91. Saratoga Springs, N. Y	8,000,000
92. Springfield, Ohio	5,000,000
93. Olean, N. Y	2,000,000
94. Dayton, Ohio	10,000,000
95. Urbana, Ohio	2,000,000
96. Brantford, Canada	2,000,000
97. Mankato, Minn	1,500,000
98. Wichita, Kan	5,000,000
99. East Saginaw, Mich	12,000,000
100. South Denver, Colo	2,000,000
101. Pueblo, Colo	5,500,000
102. Pueblo, Colo	5,500,000

Place.	Capacity.
103. Schenectady, N. Y	5,000,000
104. Indianapolis, Ind	12,000,000
105. Bucyrus, Ohio	1,500,000
106. Portland, Ore	12,000,000
107. Binghamton, N. Y.	12,000,000
108. Omaha, Neb	3,000,000
109. Anniston, Ala	3,000,000
110. Anniston, Ala	3,000,000
111. Denver, Colo	6,000,000
112. Denver, Colo.	6,000,000
113. Kalamazoo, Mich	3,000,000
114. Terre Haute, Ind	6,000,000
115. Covington, Ky.	5,000,000
116. Covington, Ky	5,000,000
117. Emporia, Kan	3,000,000
118. Norfolk, Va	7,000,000
119. Fresno, Cal	4,000,000
120. West New Brighton, N. Y.	5,000,000
121. Galesburg, Ill	1,500,000
122. Galesburg, Ill	1,500,000
123. Marquette, Mich	3,000,000
124. St. Joseph, Mo	6,000,000
125. Rockford, Ill	6,000,000
126. Duluth, Minn	5,000,000
127. Taunton, Mass	4,000,000
128. East St. Louis, Ill	6,000,000
129. Portland, Ore	2,500,000
130. Batavia, N. Y	2,500,000
131. Cheyenne, Wyo	3,000,000
132. Burlington, Iowa	6,000,000
133. Cincinnati, Ohio	6,000,000
134. Cincinnati, Ohio	6,000,000
135. World's Columbian Exposition	12,000,000
136. World's Columbian Exposition	12,000,000
137. Atlanta, Ga	10,000,000
138. Atlanta, Ga	10,000,000
139. Atlanta, Ga	10,000,000
140. Wilkesbarre, Pa	5,000,000
141. Buffalo, N. Y	20,000,000

Total capacity (nominal) .... 702,000,000 gallons daily.
Total capacity (100 feet head) .... 1,400,150,000 gallons daily.

The Holly Manufacturing Company has in the course of its business built Pumping Engines of different types as follows, making in all 400 Pumping Engines of a total nominal capacity of 1,256,120,000 gallons daily; Average per Engine, 3,140,400 gallons; Average Head, 185 1-2 feet. Total daily capacity, 100 feet head, 2,330,102,600 gallons.

ALABAMA.

Name of Place.	Style of Machinery.	Capacity.
Anniston	Gaskill Horizontal	3,000,000
Anniston	Gaskill Horizontal	3,000,000
Mobile	Gaskill Horizontal	6,000,000
Mobile	Gaskill Horizontal	6,000,000

CALIFORNIA.

Fresno	Gaskill Horizontal	4,000,000
Fresno	Gaskill Duplex	1,000,000
Sacramento	Holly Quadruplex	3,000,000
San Diego	Gaskill Horizontal	2,000,000
San Diego	Gaskill Horizontal	3,000,000

COLORADO.

Colorado Springs	Holly Quadruplex	1,000,000
Denver	Gaskill Water Power	2,000,000
Denver	Gaskill Water Power	2,000,000
Denver	Gaskill Water Power	5,000,000
Denver	Holly	2,000,000
Denver	Gaskill Horizontal	6,000,000
Denver	Gaskill Horizontal	6,000,000
Denver	Gaskill Steam Power Engine	....
Denver	Gaskill Duplex	4,000,000
Fort Collins	Gaskill Water Power	750,000
Fort Collins	Gaskill Water Power	750,000

NAME OF PLACE.	STYLE OF MACHINERY.	CAPACITY.
Golden	Holly Quadruplex	1,500,000
Greeley	Gaskill Duplex	1,000,000
Highlands	Gaskill Duplex	700,000
Pueblo	Holly Quadruplex	1,500,000
Pueblo	Gaskill Horizontal	1,500,000
Pueblo	Gaskill Horizontal	5,500,000
Pueblo	Gaskill Horizontal	5,500,000
Silver Cliff	Holly Quadruplex	1,500,000
South Denver	Gaskill Horizontal	2,000,000
	DAKOTA.	
Fargo	Holly Quadruplex	1,500,000
Fargo	Gaskill Duplex	750,000
Fort Yates	Gaskill Duplex	200,000
	DELAWARE.	
Dover	Gaskill Duplex	1,000,000
Newark	Gaskill Duplex	500,000
Newark	Gaskill Duplex	500,000
	DISTRICT OF COLUMBIA.	
Washington	Gaskill Horizontal	2,500,000
Washington	Gaskill Horizontal	2,500,000
	GEORGIA.	
Atlanta	Holly Quadruplex	3,000,000
Atlanta	Holly Quadruplex	4,000,000
Atlanta	Gaskill Horizontal	10,000,000
Atlanta	Gaskill Horizontal	10,000,006
Atlanta	Gaskill Horizontal	10,000,000
	ILLINOIS.	
Chicago	Gaskill Horizontal	12,000,000
Chicago	Gaskill Horizontal	12,000,000
Columbian World's Exposition	Gaskill Horizontal	12,000,000
Columbian World's Exposition	Gaskill Horizontal	12,000,000
Decatur	Holly Quadruplex	1,000,000
East St. Louis	Gaskill Horizontal	6,000,000
Evanston	Holly Quadruplex	2,000,000
Evanston	Gaskill Horizontal	5,000,000
Galesburg	Gaskill Horizontal	1,500,000
Galesburg	Gaskill Horizontal	1,500,000
Hyde Park	Holly Quadruplex	2,000,000
Hyde Park	Gaskill Horizontal	12,000,000
Joliet	Gaskill Horizontal	3,000,000
Lake View	Gaskill Horizontal	12,000,000
LaSalle	Gaskill Horizontal	2,000,000
Litchfield	Holly Quadruplex	750,000
McCormick Harvester Company, Chicago, Ill	Gaskill Duplex	1,800,000
McCormick Harvester Company, Chicago, Ill	Gaskill Duplex	1,800,000
Monmouth	Gaskill Duplex	1,000,000
Nelson Knitting Company	Gaskill Water Power	1,250,000
Rock Island	Holly Quadruplex	2,000,000
Rock Island	Holly Quadruplex	3,000,000
Rockford	Holly Quadruplex	2,000,000
Rockford	Gaskill Horizontal	3,000,000

NAME OF PLACE.	STYLE OF MACHINERY.	CAPACITY.
Rockford	Gaskill Horizontal	6,000,000
Streator	Gaskill Horizontal	1,500,000
Streator	Gaskill Horizontal	1,500,000
Taylorville	Gaskill Duplex	750,000
Town of Lake	Gaskill Vertical	4,000,000
Town of Lake	Gaskill Vertical	4,000,000
Town of Lake	Gaskill Horizontal	12,000,000
	INDIANA.	
Bluffton	Gaskill Duplex	1,000,000
Columbus	Holly Quadruplex	1,500,000
Columbus	Gaskill Horizontal	3,000,000
Connersville	Holly Quadruplex	1,000,000
Evansville	Gaskill Vertical	4,000,000
Evansville	Gaskill Vertical	4,000,000
Fort Wayne	Holly Quadruplex	3,000,000
Fort Wayne	Horizontal Duplex	2,000,000
Fort Wayne	Gaskill Triple Compound	6,000,000
Indianapolis	Holly Quadruplex	6,000,000
Indianapolis	Gaskill Water Power	5,000.000
Indianapolis	Gaskill Water Power	5,000,000
Indianapolis	Gaskill Steam Power Engine	....
Indianapolis	Gaskill Steam Power Engine	....
Indianapolis	Gaskill Horizontal	12,000,000
Jeffersonville	Gaskill Vertical	1,500,000
Jeffersonville	Gaskill Vertical	1,500,000
Laporte	Holly	2,000,000
Michigan City	Gaskill Duplex	2,000,000
New Castle	Gaskill Duplex	750,000
New Castle	Gaskill Duplex	750,000
Plymouth	Gaskill Duplex	750,000
Plymouth	Gaskill Duplex	750,000
Richmond	Gaskill Horizontal	4,000,000
Terre Haute	Gaskill Vertical	6,000,000
Valparaiso	Gaskill Horizontal	1,500,000
Wabash	Gaskill Duplex	1,500,000
Wabash	Gaskill Duplex	1,500,000
Washington	Gaskill Duplex	1,500,000
Washington	Gaskill Duplex	1,000,000
	IOWA.	
Atlantic	Holly Qaudruplex	1,500,000
Atlantic	Holly Quadruplex	1,000,000
Burlington	Holly Quadruplex	3,000,000
Burlington	Gaskill Horizontal	3,500,000
Burlington	Gaskill Horizontal	6,000,000
Des Moines	Gaskill Horizontal	5,000,000
Iowa City	Holly Quadruplex	1,500,000
Iowa City	Gaskill Duplex	1,000,000
Keokuk	Holly Quadruplex	1,500,000
Le Mars	Gaskill Duplex	500,000
Le Mars	Gaskill Duplex	500,000
Oskaloosa	Holly Quadruplex	1,500,000
Oskaloosa	Holly Quadruplex	1.000,000
Sioux City	Gaskill Horizontal	3,000,000
Sioux City	Gaskill Horizontal	3,000,000

KANSAS.

Name of Place.	Style of Machinery.	Capacity.
Abilene	Gaskill Duplex	1,000,000
Clay Center	Gaskill Duplex	1,500,000
Emporia	Holly Quadruplex	1,500,000
Emporia	Gaskill Vertical	2,000,000
Hutchinson	Gaskill Duplex	1,500,000
Hutchinson	Gaskill Horizontal	1,500,000
Leavenworth	Gaskill Horizontal	5,000,000
Olathe	Gaskill Duplex	750,000
Salina	Gaskill Horizontal	1,500,000
Topeka	Holly Quadruplex	1,500,000
Topeka	Holly Quadruplex	1,500,000
Topeka	Gaskill Horizontal	4,000,000
Wichita	Gaskill Horizontal	5,000,000

KENTUCKY.

Covington	Holly	4,000,000
Covington	Gaskill Vertical	5,000,000
Covington	Gaskill Vertical	5,000,000
Frankfort	Gaskill Vertical	2,000,000
Lexington	Gaskill Horizontal	1,500,000
Lexington	Gaskill Horizontal	1,500,000
Louisville	Gaskill Duplex	3,000,000
Newport	Gaskill Vertical	4,000,000
Newport	Gaskill Vertical	4,000,000
Owensboro	Gaskill Vertical	2,000,000
Paducah	Gaskill Vertical	2,000,000
Paducah	Gaskill Vertical	2,000,000

MAINE.

Augusta	Gaskill Water Power	3,000,000
Bangor	Holly	4,000,000
Bangor	Gaskill Water Power	5,000,000
Rockland	Holly	1,000,000
Winter Harbor	Gaskill Duplex	300,000

MARYLAND.

Cumberland	Holly	1,500,000
Cumberland	Holly Quadruplex	2,000,000
Easton	Gaskill Duplex	500,000
Easton	Gaskill Duplex	250,000

MASSACHUSETTS.

Beverly	Gaskill Horizontal	2,000,000
Beverly	Gaskill Horizontal	2,000,000
Boston	Gaskill Horizontal	8,000,000
Boston	Gaskill Horizontal	8,000,000
Milford	Holly	1,500,000
Randolph	Gaskill Duplex	2,500,000
Taunton	Holly Quadruplex	2,000,000
Taunton	Holly	2,000,000
Taunton	Gaskill Horizontal	4,000,000

MICHIGAN.

NAME OF PLACE	STYLE OF MACHINERY.	CAPACITY.
Adrian	Gaskill Horizontal	1,500,000
Adrian	Gaskill Horizontal	1,500,000
Allegan	Holly	1,000,000
Alpena	Holly	2,000,000
Bay City	Holly Quadruplex	2,000,000
Bay City	Gaskill Horizontal	5,000,000
Big Rapids	Holly	1,000,000
Big Rapids	Gaskill Water Power	1,500,000
Big Rapids	Gaskill Water Power	1,500,000
Cadillac	Gaskill Duplex	2,000,000
Dowagiac	Gaskill Duplex	1,000,000
Dowagiac	Gaskill Duplex	1,000,000
East Saginaw	Holly Quadruplex	2,000,000
East Saginaw	Holly Quadruplex	6,000,000
East Saginaw	Gaskill Horizontal	12,000,000
East Tawas	Gaskill Duplex	750,000
East Tawas	Gaskill Duplex	750,000
Flint	Gaskill Horizontal	1,500,000
Flint	Gaskill Horizontal	1,500,000
Flint	Gaskill Duplex	4,000,000
Jackson	Holly Quadruplex	1,500,000
Jackson	Gaskill Horizontal	4,000,000
Kalamazoo	Holly Quadruplex	2,000,000
Kalamazoo	Gaskill Vertical	3,000,000
Kalamazoo	Gaskill Vertical	3,000,000
Ludington	Gaskill Duplex	3,000,000
Marquette	Holly	1,500,000
Marquette	Gaskill Horizontal	3,000,000
Manistee	Holly Quadruplex	2,000,000
Manistee	Gaskill Duplex	750,000
Muskegon	Gaskill Vertical	3,000,000
Muskegon	Gaskill Vertical	3,000,000
Port Huron	Holly Quadruplex	2,000,000
Port Huron	Gaskill Horizontal	5,000,000
Saginaw City	Holly Quadruplex	2,000,000
Saginaw City	Gaskill Horizontal	4,000,000
Vassar	Gaskill Duplex	1,500,000
West Bay City	Gaskill Horizontal	1,500,000
West Bay City	Gaskill Horizontal	1,500,000

MINNESOTA.

Duluth	Gaskill Horizontal	5,000,000

MISSOURI.

Carrollton	Gaskill Duplex	1,000,000
Kansas City	Holly Quadruplex	3,000,000
Kansas City	Holly	1,500,000
Kansas City		7,000,000
Kansas City	Holly Quadruplex	4,000,000
Kansas City	Gaskill Horizontal	4,000,000
Kansas City	Gaskill Vertical Pumps, Horizontal Engine,	10,000,000
Kansas City	Gaskill Vertical Pumps, Horizontal Engine,	10,000,000
Kansas City	Gaskill Horizontal	9,000,000
Sedalia	Holly	1,500,000
St. Joseph	Gaskill Horizontal	6,000,000

NEBRASKA.

Name of Place.	Style of Machinery.	Capacity
Beatrice	Gaskill Horizontal	1,500,000
Beatrice	Gaskill Horizontal	1,500,000
Chadron	Gaskill Duplex	1,000,000
Chadron	Gaskill Duplex	500,000
Omaha	Gaskill Horizontal	5,000,000
Omaha	Gaskill Horizontal	14,000,000
Omaha	Gaskill Horizontal	3,000,000
Omaha	Gaskill Triple Compound	8,000,000
Seward	Gaskill Duplex	1,000,000

NEW HAMPSHIRE.

Nashua	Gaskill Water Power	3,000,000

NEW JERSEY.

Asbury Park	Gaskill Duplex	750,000
Asbury Park	Gaskill Duplex	500,000
Asbury Park	Gaskill Duplex	500,000
Atlantic City	Gaskill Duplex	4,000,000
Burlington	Gaskill Horizontal	1,500,000
East Orange	Gaskill Horizontal	1,500,000
Merchantville	Gaskill Duplex	500,000
Merchantville	Gaskill Duplex	500,000
Salem	Holly Quadruplex	1,500,000

NEW MEXICO.

Albuquerque	Gaskill Horizontal	2,000,000

NEW YORK.

Auburn	Holly Quadruplex	4,000,000
Auburn	Gaskill Horizontal	7,500,000
Auburn	Gaskill Horizontal	7,500,000
Batavia	Gaskill Duplex	1,500,000
Batavia	Gaskill Horizontal	1,500,000
Batavia	Gaskill Horizontal	2,500,000
Binghamton	Holly Quadruplex	2,000,000
Binghamton	Holly Quadruplex	6,000,000
Binghamton	Holly Quadruplex	1,000,000
Binghamton	Gaskill Horizontal	12,000,000
Buffalo	Holly Quadruplex	4,000,000
Buffalo	Holly Quadruplex	6,000,000
Buffalo	Gaskill Horizontal	15,000,000
Buffalo	Gaskill Horizontal	20,000,000
Buffalo	Gaskill Horizontal	20,000,000
Buffalo	Gaskill Horizontal	20,000,000
Chatham	Gaskill Duplex	1,000,000
Dunkirk	Holly Quadruplex	2,000,000
East Aurora	Gaskill Water Power	700,000
Flushing	Holly Quadruplex	1,500,000
Fort Niagara	Gaskill Duplex	500,000
Garden City	Holly Quadruplex	1,500,000
Geneseo	Gaskill Duplex	600,000
Gouverneur	Holly Quadruplex	1,500,000
Haverstraw	Gaskill Duplex	500,000
Jackson Lumber Company	Gaskill Duplex	320,000
Jamaica	Gaskill Duplex	1,000,000

Name of Place.	Style of Machinery.	Capacity.
Jamaica	Gaskill Duplex	1,000,000
Lockport	Holly Water Power	2,000,000
Lockport	Gaskill Water Power	3,000,000
Lockport	Gaskill Water Power	5,000,000
Long Island City	Holly Quadruplex	3,000,000
Mamaroneck	Gaskill Water Power	250,000
Middletown	Holly Water Power	2,000,000
New Brighton	Gaskill Horizontal	5,000,000
Ogdensburgh	Holly Water Power	2,000,000
Olean	Gaskill Horizontal	2,000,000
Patchogue	Gaskill Duplex	700,000
Patchogue	Gaskill Duplex	700,000
Potsdam	Holly Water Power	1,500,000
Pulaski	Gaskill Water Power	1,000,000
Rochester	Holly Water Power	2,000,000
Rochester	Holly Water Power	2,000,000
Rochester	Holly Quadruplex	3,000,000
Rochester	Gaskill Horizontal	2,000,000
Rockaway Beach	Holly Quadruplex	1,500,000
Saratoga Springs	Holly Quadruplex	2,000,000
Saratoga Springs	Gaskill Horizontal	5,000,000
Saratoga Springs	Gaskill Horizontal	8,000,000
Schenectady	Holly	2,000,000
Schenectady	Gaskill Horizontal	3,000,000
Schenectady	Gaskill Horizontal	5,000,000
Springville	Gaskill Water Power	1,000,000
Suspension Bridge	Holly Water Power	1,500,000
Suspension Bridge	Gaskill Water Power	5,000,000
Tonawanda	Gaskill Horizontal	2,000,000
Tonawanda	Gaskill Horizontal	2,000,000
Troy	Gaskill Quadruplex	6,000,000
Troy	Gaskill Quadruplex	6,000,000

OHIO.

Name of Place.	Style of Machinery.	Capacity.
Bucyrus	Gaskill Horizontal	1,500,000
Canton	Holly Water Power	2,000,000
Cincinnati	Gaskill Horizontal	6,000,000
Cincinnati	Gaskill Horizontal	6,000,000
Circleville	Gaskill Duplex	1,500,000
Circleville	Gaskill Duplex	1,500,000
Columbus	Holly Quadruplex	3,000,000
Columbus	Holly Quadruplex	4,000,000
Columbus	Gaskill Horizontal	10,000,000
Columbus	Gaskill Triple Compound	7,500,000
Columbus	Gaskill Triple Compound	7,500,000
Dayton	Holly Quadruplex	4,000,000
Dayton	Holly Quadruplex	4,000,000
Dayton	Gaskill Horizontal	10,000,000
Franklin	Gaskill Duplex	750,000
Franklin	Gaskill Duplex	750,000
Ironton	Holly	2,000,000
Kent	Gaskill Duplex	1,000,000
Kent	Gaskill Duplex	1,000,000
Kenton	Holly Quadruplex	1,500,000

Name of Place.	Style of Machinery.	Capacity.
Kenton	Gaskill Duplex	1,000,000
Lima	Gaskill Horizontal	3,000,000
Lima	Gaskill Horizontal	3,000,000
Mansfield	Holly	2,000,000
Middletown	Holly Quadruplex	2,000,000
Norwalk	Holly	2,000,000
Portsmouth	Holly	2,000,000
Sidney	Holly	2,000,000
Springfield	Gaskill Horizontal	5,000,000
Tiffin	Holly Quadruplex	1,500,000
Tiffin	Gaskill Horizontal	3,000,000
Urbana	Holly Quadruplex	1,500,000
Urbana	Gaskill Horizontal	2,000,000
Van Wert	Holly Duplex	1,500,000
Van Wert	Holly Duplex	1,500,000
Youngstown	Holly	2,000,000

OREGON.

Enterprise	Gaskill Water Power	700,000
Portland	Gaskill Horizontal	1,500,000
Portland	Gaskill Horizontal	12,000,000
Portland	Gaskill Horizontal	2,500,000

PENNSYLVANIA.

Beaver Falls	Gaskill Water Power	2,000,000
Beaver Falls	Gaskill Water Power	2,000,000
Beaver Falls	Holly Quadruplex	1,500,000
Bloomsburg	Gaskill Duplex	2,500,000
Butler	Gaskill Duplex	2,000,000
Chester	Gaskill Horizontal	4,000,000
Chester	Gaskill Horizontal	4,000,000
City Hall, Philadelphia	Gaskill Duplex	1,000,000
City Hall, Philadelphia	Gaskill Duplex	200,000
Corry	Gaskill Duplex	1,000,000
Corry	Gaskill Duplex	1,000,000
Danville	Holly Quadruplex	1,500,000
Doylestown	Gaskill Horizontal	1,500,000
Erie	Gaskill Horizontal	5,000,000
Holmesburg	Gaskill Duplex	1,000,000
Monongahela City	Gaskill Duplex	1,000,000
Oil City	Gaskill Duplex	1,500,000
Philadelphia	Gaskill Horizontal	20,000,000
Philipsburg	Gaskill Duplex	1,000,000
Titusville	Holly Quadruplex	2,000,000
Towanda	Gaskill Duplex	1,000,000
Union City	Gaskill Duplex	750,000
Union City	Gaskill Duplex	750,000
Wilkesbarre	Gaskill Horizontal	5,000,000

TENNESSEE.

Memphis	Holly	4,000,000
Memphis	Gaskill Vertical	4,000,000
Memphis	Gaskill Vertical	4,000,000
Nashville	Gaskill Vertical	10,000,000

TEXAS.

Name of Place.	Style of Machinery.	Capacity.
Austin	Holly Quadruplex	3,000,000
Austin	Gaskill Horizontal	5,000,000
Dallas	Gaskill Horizontal	8,000,000
Denison	Gaskill Horizontal	1,500,000
Denison	Gaskill Horizontal	1,500,000
Fort Worth	Holly Quadruplex	3,000,000
Gainesville	Gaskill Horizontal	1,500,000

VERMONT.

Vergennes	Holly	1,000,000

VIRGINIA.

Lynchburg	Gaskill Water Power	3,000,000
Norfolk	Holly Quadruplex	2,000,000
Norfolk	Gaskill Horizontal	2,000,000
Norfolk	Gaskill Horizontal	5,000,000
Norfolk	Gaskill Vertical	7,000,000
Portsmouth	Gaskill Water Power	1,750,000
Portsmouth	Gaskill Water Power	600,000
Radford	Gaskill Duplex	1,000,000
Richlands	Gaskill Duplex	1,000,000
Roanoke	Gaskill Duplex	2,000,000
Roanoke	Gaskill Duplex	2,000,000

WASHINGTON.

New Tacoma	Gaskill Water Power	1,000,000
North Yakima	Gaskill Water Power	1,250,000
North Yakima	Gaskill Water Power	1,250,000
Spokane Falls	Gaskill Water Power	750,000
Spokane Falls	Gaskill Water Power	750,000
Spokane Falls	Gaskill Water Power	3,200,000
Spokane Falls	Gaskill Water Power	4,600,000
Tacoma	Gaskill Water Motor	1,500,000

WEST VIRGINIA.

Martinsburg	Holly	1,500,000

WISCONSIN.

Ashland	Gaskill Horizontal	1,500,000
Ashland	Gaskill Horizontal	1,500,000
Fond du Lac	Gaskill Horizontal	3,000,000
Fond du Lac	Gaskill Horizontal	3,000,000
Oshkosh	Gaskill Horizontal	4,000,000
Oshkosh	Gaskill Horizontal	4,000,000
Stevens Point	Gaskill Duplex	2,000,000
Stevens Point	Gaskill Duplex	2,000,000
Wausau	Gaskill Horizontal	3,000,000
Wausau	Gaskill Horizontal	3,000,000

WYOMING.

Cheyenne	Gaskill Horizontal	3,000,000

CANADA.

NAME OF PLACE.	STYLE OF MACHINERY.	CAPACITY.
Brantford	Gaskill Horizontal	2,000,000
Brockville	Gaskill Horizontal	1,500,000
Brockville	Gaskill Horizontal	1,500,000
Fredericton	Gaskill Horizontal	1,500,000
Ste. Cunegondo	Holly Quadruplex	1,500,000

Total Number of Pumping Engines built by the Holly Manufacturing Company to date	400
Total Daily Capacity of said Engines, Nominal, Gallons	1,256,120,000
Average Daily Capacity, Nominal, Gallons	3,140,400
Average Head, Feet	185½
Total Daily Capacity, 100 feet Head, Gallons	2,330,102,600

THE FOLLOWING OFFICIAL REPORTS

of duty trials of the Gaskill-Holly High Duty Pumping Engine, all of which were made by o under direct scrutiny of disinterested experts and practical engineers, makes a record which no other pumping engine can equal.

HARVEY F. GASKILL.

THE Pumping Engine of which this book is the subject takes its name from the designer and inventor, HARVEY F. GASKILL, who died on the first day of April, 1889, at the age of forty-four years, from a complication of diseases affecting the stomach, after an illness of six months.

The following resolutions are here given as a token of the respect and esteem in which he was held by the stockholders and officers of the Company:

At a meeting of the stockholders of the Holly Manufacturing Company, April 2, 1889, to take appropriate action respecting the demise of HARVEY F. GASKILL, the late Vice-President and Engineer of the Company, the following preamble and resolutions were adopted:

WHEREAS, Death has suddenly taken from our midst MR. HARVEY F. GASKILL, one of the Trustees and the Vice-President of the Company; therefore,

Resolved, That in this event we mourn the loss of one who for sixteen years has been a member of this Company, and one of our number, and whose counsel and advice was wise and judicious, whose skill and ability as a mechanical engineer was remarkable and brilliant, and whose whole life was evenly balanced, and marked with rigid integrity, justice and fairness.

Resolved, That we sincerely mourn the loss of a most valuable and able colaborer, as a Trustee, Vice-President, Superintendent, and Mechanical Engineer of this Company, and express our high appreciation of the able and faithful manner in which he has ever discharged the duties of each and all these positions, and all other relations which he has sustained to the Company, and each member of it individually.

Resolved, That in his death business circles of this city and society at large have suffered a great loss: the church with which he was connected a consistent, valued and trusted member; and mechanical science a brilliant, useful, and successful inventor.

Resolved, That we extend to the family of the deceased our heartfelt sympathy in their bereavement.

Resolved, That the stockholders of this Company attend the funeral of the deceased in a body, and that an engrossed copy of these resolutions be delivered to the family.

THE COLUMBIAN WORLD'S EXPOSITION

AND THE

GASKILL HIGH DUTY PUMPING ENGINE.

The City of Chicago recently entered into contract with this Company for two of the Gaskill horizontal, compound, condensing, high duty pumping engines, each of 12,000,000 gallons daily capacity against 90 pounds, equivalent to a head of about 208 feet water pressure, and six horizontal tubular steam boilers.

The plant will be erected at the Sixty-eighth Street Water Works of the city (which street is next to the southern boundary of the Exposition Grounds), near the shore of Lake Michigan.

Water will be conveyed to the pumps through a capacious tunnel extending from the station to a cribbed inlet situated about two miles out in the lake.

The ordinary city water pressure is not sufficient for the requirements of exhibitors of hydraulic and hydrostatic apparatus, such as elevators, motors, presses, etc., and these engines will be provided, independently of the regular city service, with large iron water mains running in and through the Exposition Grounds, from which branching pipes of less diameter will convey water to the different buildings, fountains, etc. This independent high pressure service will insure a constant and abundant supply of water to the most elevated parts of the grounds and buildings, at all hours, day and night.

After the close of the Exposition the engines will be used to supplement the present city water supply.

The engines and boilers are guaranteed to show a duty equal to raising 110,000,000 pounds of water one foot high with 100 pounds of coal.

An engine of the same size and kind heretofore erected at this station will be continued in use; and another of the same pattern now at the Town of Lake pumping station will be removed to 68th Street, so that there will be four of these engines in operation at the latter place, having an aggregate capacity of 48,000,000 gallons daily.

Two more are in use at the North Pumping Works, and yet another at the Lake View Pumping Works, making in all **seven high duty Gaskill pumping engines** purchased by the **City of Chicago within six years**, having a capacity to pump 3,500,000 gallons per hour, or 84,000,000 gallons of water per 24 hours.

Official report of duty trials of the engines at the North Pumping Works is given on pages 124–133, and of the Town of Lake engine on pages 180–190, this book.

Every facility will be afforded visitors for as thorough inspection of the engines at 68th Street as they may desire.

SARATOGA SPRINGS, N. Y.

NOVEMBER, 1882.

REPORT OF

JOHN W. HILL, M. E., CINCINNATI, OHIO, AND PROF. DAVID M. GREENE, DIRECTOR RENSSELAER POLYTECHNIC INSTITUTE, TROY, N. Y.

To the Honorable Board of Water Commissioners, HIRAM OWEN, MICHAEL WALSH, DAVIS COLEMAN, HENRY B. HANSOM *and* BENJAMIN W. CLAPP, *Saratoga, N. Y.*

GENTLEMEN—Acting respectively for and in behalf of your Board and of the contractors, we have made a careful examination of the Gaskill Compound Pumping Engine recently erected for your public water service by the Holly Manufacturing Company of Lockport, N. Y., and beg to report thereon as follows :

The engine is horizontal, of the rotative non-receiver compound beam type, and involves a novel arrangement of the several members, whereby a large capacity and a high economy are obtained by a simple machine in a small compass. The contract provides that the engine shall "have four steam cylinders arranged in pairs; one high pressure cylinder, and one low pressure cylinder in each pair, the low pressure cylinder placed under the high pressure cylinder. There will be one pump to each pair of steam cylinders. The whole bolted to a heavy cast iron bed-plate which is bolted in turn to stone foundations. The fly wheel will revolve between the two pump cylinders. The crank shaft to have its pillow blocks one on each pump cylinder. The cross heads of the high pressure cylinders will be (are) connected by suitable links to a beam which will be (is) in turn connected, one end to the crank and the other end to the piston rod of the low pressure cylinders and pumps. The supports for the beam will rest on and be firmly bolted to the base."

"The steam cylinders will be four in number; two high pressure (cylinders) 21 inches diameter and 36 inches stroke of pistons, and two low pressure (cylinders) 42 inches diameter and 36 inches stroke of pistons; to be steam jacketed on sides and heads."

"The pumps will be two in number; and will be of the kind known as double-acting plunger pumps; the plungers of which will be 20 inches diameter and 36 inches stroke; and will be packed with an internal gland, which will be adjusted by means of bolts passing through the rear of the pumps, and will be made easy of access by means of hand holes placed in the pump cylinders "

"The pump piston rods will be two in number; one for each pump, and will be 4 inches diameter."

The contract provides that the engine shall have a "pumping capacity of four million U. S. gallons in twenty-four hours against a domestic pressure of eighty pounds per square inch, at eighteen revolutions per minute," and shall "develop a duty the equivalent of eighty million pounds of water raised one foot high with a consumption of one hundred pounds of best coal."

The contract further provides that the engine shall, with safety to all its parts, "carry a fire pressure of one hundred and forty pounds per square inch," and shall be capable "of working safely at thirty revolutions per minute," and shall have an "average daily duty of sixty-five million foot-pounds with good merchantable anthracite coal, said average daily duty to be determined by the engineer at the works, who shall keep a daily record of the coal consumed and the work performed by the engines and pumps."

Omitting the daily average duty of "sixty-five millions," which, by the terms of contract, is not within the purview of our investigation, we have endeavored to ascertain the compliance of the engine and boilers (the contractors having furnished the entire plant) with the following requirements of the contract:

First. A delivery into the mains at eighty pounds pressure, of four million standard gallons of water, the engine working at a speed not in excess of eighteen revolutions per minute.

The plunger diameter and strokes were carefully measured and found to be exactly in accordance with the specified dimensions, or 20 inches diameter and 36 inches stroke, with rods 4 inches diameter.

The calculated delivery of two such plungers per double stroke or per revolution of engine is 191.923 gallons, of which quantity we estimate from previous experiments upon pumps in all material respects like these, 97.5 per cent. will be actually delivered to the force pipe, or 187.125 gallons per revolution, and at 18 revolutions per minute. for 24 hours, an actual delivery of

187.125 × 18 × 60 × 24 = 4,850,280 gallons, or an excess of 21.257 per cent. over contractor's guarantee.

During the duty trial of Nov. 1st and 2d the engine ran at an average speed of 17.8742 revolutions per minute, corresponding to an actual daily delivery, with an estimated slip or loss of action of 2.5 per cent., of 4,816,371.02 gallons against an average observed pressure upon contractors' gauge and special test gauge of 95.07 pounds per square inch. It is scarcely necessary to suggest that if the engine is capable of pumping the above quantity of water against a pressure of 95 pounds per square inch that it is capable of pumping a larger quantity against a reduced pressure, say 80 pounds, with the same boiler pressure.

The pumps, by reason of the very low speed at which they are now worked, are supplied with ordinary by-pass pipes and valves to adapt the water load to the reduced motion of engine.

During the trial of Nov. 1st and 2d the by-passes were closed and pumps required to furnish a full delivery of water.

Second. A duty of eighty million pounds of water raised one foot high with a consumption of one hundred pounds of best coal.

The duty trial began at 8.00 A. M., Nov. 1st, and terminated at 4.00 A. M., Nov. 2d, covering an unbroken run of twenty hours, during which time at regular intervals observations were made of the engine counter, steam and water pressure gauges, water levels in boilers, temperatures of injection, overflow water to heater, and feed water to boilers, meter in feed pipe, and vacuum gauge.

The coal burned was "Lackawanna," of excellent quality, as shown by the remarkably small percentage of refuse during the trial. This was weighed in double charges of 150 pounds each and dumped into the coal wagon in front of the boilers; each charge to the boilers of 300 pounds was exhausted before the next charge was permitted in the wagon.

Indicator diagrams from all the steam cylinders were taken at irregular intervals during the trial.

From the log of trial are obtained the following averages and totals as affecting the duty:

DATA FROM DUTY TRIAL.

Engine counter at 8.00 A. M., Nov. 1st	286,053
" " " 4.00 A. M., Nov. 2d	307,502
Revolutions for 20 hours	21,449
Average pressure by water gauge	95.06875
" " engine steam gauge	74.25
" " vacuum gauge	27.295
" temperature of injection	56.225
" " overflow	71.125
Total coal burned, pounds	6,750

From which data we deduce the duty as follows: First, by the generally adopted method of

$$D = \frac{A \times P \times F \times 100}{C}$$

Where A represents the mean area of one plunger at right angles to its axis; P represents the total pressure per square inch of plunger, and consists, first, of the observed pressure by gauge, × or — difference of levels (center of water pressure gauge and source of supply), × an allowance for extra frictional resistances of water passages into and out of pumps usually taken at one pound, and F represents the total plunger travel during trial, in feet. C represents the total coal burned during trial. The condition of fires being alike at beginning and end of trial, with minimum fluctuations of steam pressure and water levels in boilers; and second as a net or absolute duty by the method

$$D = \frac{G \times W \times H \times 100}{C}$$

Where G represents the actual delivery of water during trial in U. S. standard gallons; W the weight of water at observed temperature per gallon; and H the head in feet through which the water is pumped, consisting of the algebraic sum of the head due pressure by gauge, and the differences of levels (center of water pressure gauge and source of supply); C, as before, represents the total coal burned during trial.

Calculating duty by first method

$$A = \frac{20^2 \times .7854 \times (20^2 \times .7854 - 4^2 \times .7854)}{2} = \dots\dots\dots \qquad 307.8768 \text{ sq. inches.}$$

P = pressure by gauge mean of 81 readings	95.06875	
Difference levels center of water pressure gauge and source of supply $\left(\frac{.23}{2.308}\right)$	0.09965	
Allowance for extra frictional resistances of water passages into and out of pumps	1.00000	96.1684 pounds.
F = revolutions during trial	21,449	
Piston travel per revolution	12	257,388 feet.
C = total coal burned		6,750 pounds.

then by formula

$$\frac{307.8768 \times 96.1684 \times 257,388 \times 100}{6,750} = \mathbf{112,899,983.104}$$ **foot-pounds** duty, or the work of raising nearly one hundred and thirteen million pounds of water one foot high, without frictional resistance or loss of effect.

The contract provides for a duty of eighty millions, which is exceeded by the duty based upon the conventional formula, by 41.125 per cent.

Calculating the duty by second method

G = 191.923 × .975 × 21,449 = ... 4,013,642.516 gallons.
W = weight per gallon taken for temperature, 56............. 8.33 pounds.
H = pressure by gauge 95.06875 × 2.308..................... 219.418
Difference levels center of gauge and source of supply......... 0.230 219.648 feet.
C = total coal burned .... 6,750 pounds.

then by formula

$$\frac{4{,}013{,}642.516 \times 8.33 \times 219.648 \times 100}{6{,}750} = 108{,}793{,}535.3$$ pounds of water raised one foot high as the equivalent of absolute duty.

The contract provides for a duty of eighty millions, which is exceeded by the net duty nearly 36 per cent.

In the following table are recapitulated the averages of all data and totals for duty trial:

Duration of trial, hours..................................	20
Average pressure by engine, gauge, pounds	74.25
" vacuum formed, inches	27.295
" temperature of injection, Fahr....................	56.225
" " overflow, "	71.125
" " feed water to boilers..............	169.175
" pressure by gauge on force main, pounds..........	95.06875
Difference levels center of pressure gauge and source of supply, pounds..	0.09965
Revolutions in 20 hours..................................	21,449
" per minute..................................	17.8742
Piston speed per minute, feet	107.2452
Total coal burned, pounds................................	6,750
Calculated discharge of pumps per revolution, gallons.....	191.923
Actual estimated discharge of pumps per revolution, gallons,	187.125
Weight of water per gallon at 56 F., pounds	8.33
Contract delivery at 18 revolutions, gallons	4,000,000
Actual estimated delivery at 18 revolutions, gallons.... ...	4,850,280
Excess over contractors' guarantee, per cent.................	21.257
Contract duty per 100 pounds of coal	80,000,000
Conventional duty as per trial.............................	112,899,983.104
Excess over contractors' guarantee, per cent................	41.125
Net absolute duty as per trial.............................	108,793,535.3
Excess over contractors' guarantee, per cent...	36

Just before the duty trial terminated all the principal bearings of engine were examined with no evidences of heating.

Third. The furnishing of a fire pressure of 140 pounds per square inch with safety to all parts of the engine.

This is a moderate requirement, as pumping engines for fire service are usually constructed to withstand 150 pounds pressure per square inch of plunger or piston, and are not infrequently required to furnish a fire pressure of 180 to 200 pounds per square inch in towns built on undulating ground. Without special observation there is little doubt of the ability of the engine to meet this requirement with a large factor of safety; but during the test of fire streams, Saturday, October 28th, the gauge was read at frequent intervals and occasionally registered in excess of 140 pounds.

Fourth. The engine shall be capable of working safely at thirty revolutions per minute.

It is not possible to properly test the engine at thirty revolutions per minute with the present connecting mains. But from the manner in which the engine performed during the duty trial and our previous experience with similar pumping engines, there is no reasonable doubt of the dimensions and arrangement of parts being sufficient and well adapted for such a speed as a maximum requirement.

As a matter of interest to your board and ourselves, arrangements were made to determine the economy of the boilers during the duty trial, with the following results:

Duration of trial, hours	20
Average pressure by gauge, pounds	76.644
" temperature of feed water	169.175
" " water to meter	85.575
" percentage of water entrained	6.273
Water by meter record, cubic feet	1181.8
Error of meter record, percentage	2.8457
Weight of water per cubic foot (temp. 85.575), pounds	62.135
" " passed through meter	75,520.773
" " drawn off for temperature of feed, pounds	214
" " to boilers, pounds	75,306,773
" " entrained, "	4723.994
" net steam, pounds	70,582.779
" coal burned, "	6,750
Steam per pound of coal, from temperature of feed, pounds	10.4567
Steam per pound of coal from and at 212, pounds	11.286
Weight of ash and clinker returned	216
Percentage of non-combustible	3.2
Coal burned per sq. foot of grate surface, per hour	5.833
Steam per sq. foot of heating surface, per hour	1.1933

The specifications for the engine provides for "Air pumps to remove the air, water of condensation and condensing water from jet condensers," to be worked from the engine shaft. In place of which the contractor has substituted the now well-known Bulkley condensing apparatus. The merit of this change must be measured by the results. The vacuum obtained ranges from 27–28 inches, and the engine power, which with air pumps would be absorbed in discharging the contents of condensers, is now utilized in forcing water into the mains.

Whatever gain in economy of performance has been obtained by the change in condensing apparatus is a benefit alike to the contractor in his increased duty upon trial and a continuous benefit to the water service to the extent of power which otherwise would be required to work the air pumps.

The specifications provide for an automatic pressure regulator to be furnished with the engine; this, as the contractor informs us, will be supplied and attached in due time.

Barring an unevenness of some bearings (which a limited use and proper attention to working joints will soon remedy) the performance of the engine is very satisfactory.

There is a defect in the cut-off valve motion of the left engine, to which we have called the contractor's attention, and which he promises to correct. When corrected the present hesitation of the engine in turning the right inboard center will in our opinion disappear.

In conclusion, we beg to suggest as our opinion that the engine complies in all essential respects with the terms of the contract and should be accepted.

All of which is respectfully submitted.

TROY, N. Y., Nov. 3d, 1882.

JOHN W. HILL.
D. M. GREENE.

PRINCIPAL DIMENSIONS OF ENGINE.

H. P. Cylinder (2) diameter	21 inches.
L. P. Cylinder (2) "	42 "
H. P. Piston Rods (2) diameter	3 "
L. P. Piston Rods (4) "	3.5 "
Pump Plungers (2) "	20 "
" " rods (2) "	4 "
Stroke of all Pistons and Plungers	36 "
Cut off Valves	Puppet.
Intermediate Valves	Gridiron Slide.
L. P. Cylinder Exhaust Valves	" "
Fly-wheel, diameter	12.83 feet.
" weight	12,000 pounds.

PRINCIPAL DIMENSIONS OF BOILERS.

Number	Two.
Style	Horizontal return tubular.
Diameter of Shells	5.5 feet.
Length " "	18 "
Number of Tubes	87
Diameter of " outside	3 inches.
Grates (2) length	4.75 feet.
" (2) width	6 "
Heating Surface, estimated	2957.5 sq. feet.
Grate Surface	57 sq. feet.
Heating to Grate Surface, ratio	51.89

REPORT OF THE TRIAL

OF THE

GASKILL PUMPING ENGINE

AT

SARATOGA SPRINGS, N. Y., JUNE, 1883,

BY

CHARLES T. PORTER,

PUBLISHED BY

THE BOARD OF WATER COMMISSIONERS, 1883.

CORRESPONDENCE.

When the present Board of Water Commissioners organized on the first Tuesday in May, it was perfectly clear that public sentiment demanded another examination and test of our new pumping machinery, by a thoroughly competent and disinterested expert. With this purpose in view, correspondence was had with several experts in mechanical engineering, which resulted in the employment of Mr. Chas. T. Porter, of New York City. This gentleman has, we believe, made a most complete, careful, thorough and honest examination of our pumping engine. His preliminary report was received June 29th, and his complete report July 30th. Not being personally acquainted with any experts in mechanical engineering, we were obliged to be dependent on recommendations in employing a proper person to do this work. We believe it due our taxpaying citizens that they should know on whose recommendations we employed Mr. Porter. One expert who had sent us his card, referred, among others, to the Willimantic Linen Co., of Willimantic, Conn. In reply to our letter of inquiry, this Company sent us the following letter:

May 10, 1883.

MR. W. H. HODGMAN, *Water Commissioner*, Saratoga Springs, N. Y.

Dear Sir: Your favor of the 9th received. I am not sufficiently well acquainted with Mr. —— to express an opinion. He made some tests for us at one time, and this is all that I know of him. I understand that Mr. Chas. T. Porter, engineer and expert, room 42, Tribune Building, New York, is open to engagements. There is no question as to Mr. Porter's ability and knowledge of steam engines and machinery. I would rather have Mr. Porter to find out the "true inwardness" of machinery than any man I can now think of. Very truly yours,

W. E. BARROWS, *Prest.*

After receiving this letter we wrote Mr. Porter as follows:

SARATOGA SPRINGS, May 12, 1883.

MR. CHAS. T. PORTER, New York City.

Dear Sir: You have been recommended to us by the Willimantic Linen Co.'s President, as a reliable and competent steam engineer expert. We have in our village one of the new Gaskill pumping engines, of four million capacity, which has recently been put in. We are not satisfied with the test made by Messrs. Hill and Greene, and shall, in all probability, have another test made by a party or parties whom a very large majority of our people can feel have no interest in representing this engine other than what it is. We are extremely anxious to get at the exact truth, and know whether this machine is suitable to our wants, and can be depended upon for domestic as well as fire purposes. Also, as to whether the Holly Co. have fulfilled their contract with us.

The present majority in our Water Board wish to keep this machine, if they can have the assurance of some competent, unbiased expert, that it is all right and those things that now seem, to put it mildly, peculiar, explained to us. Now, I wish you would send us some references, and state what would probably be your price for this examination. I will state further that, whoever we employ, we may want them to bring their own engineer, as we prefer to have everything done by those who can have no possible interest in trying to cover any defect that may exist. You may not consider this safeguard necessary ; if so, please so state. If you happen to be consulting engineer, or are in any way connected with the manufacturers of any pumping machinery, you will do us a favor by so stating. Hoping to hear from you without delay,

I am, respectfully yours,
W. H. HODGMAN, *Water Commissioner.*

In reply to this Mr. Porter wrote as follows:

NEW YORK, May 16, 1883.

W. H. HODGMAN, *Water Commissioner*, Saratoga Springs, N. Y.

Dear Sir: I was called out of town yesterday, and find your favor of the 12th inst. on my table this morning. I take pleasure in replying to your inquiries. Until recently I have carried on the manufacture of the Porter-Allen High-Speed Steam Engine. Send you a circular that you may know what this is. Since it passed into the hands of a company I retained the direction of the manufacture until last fall. I am happy to refer you to the present management. For my position as a steam engineer and reliability, I will also refer you to two leading authorities in dynamical engineering, Prof. R. H. Thurston, of Stevens Institute, Hoboken, N. J., and Prof. Wm. P. Trowbridge, of Columbia College, this city, and to Erastus Corning, Esq., President Albany and Rensselaer Iron and Steel Co., Troy, N. Y. Probably these gentlemen will be sufficient. I have no interest whatever in any system of pumping machinery, but always feel interested for the success of any work of that character, and should hope that the result of a careful examination might be the detection and remedying of defects, rather than the rejection of the pumps, should valid grounds for dissatisfaction be found to exist. In this case all the parties are strangers to me, and I have no relations with other builders which would bias my judgment. My charge would be —— per day and my expenses, including time employed in making necessary preparations for trial and in preparing my report. Assistance in making observations and keeping records would be required. I presume that young men from the Polytechnic at Troy would be glad of the opportunity. My judgment would be against employing a special engineer. Should the result be unfavorable, the makers would be furnished with a ground of complaint. On the other hand, should defects exist the engineer could not hide them ; and if he was a sensible man he would not try. Any competent expert would *know* the complete inwardness of the machinery before he came to any conclusion respecting it.

Very truly yours, CHAS. T. PORTER.

We now wrote to Profs. Thurston and Trowbridge and Erastus Corning, Esq., asking their opinion of Mr. Porter's competency as a mechanical engineer and his reliability as a man. To these letters of inquiry we received the following replies:

TROY, N. Y., May 19, 1883.

W. H. HODGMAN, Saratoga Springs, N. Y.

Dear Sir: Yours 18th inst. received. Mr. Chas. T. Porter is a man of most excellent standing and reputation as a mechanical engineer, and we know of no man to whom we would intrust the careful examination of machinery, with the object of finding out about it, whether it has defects of any kind or character, as we would to Mr. Porter; and I do not doubt you would find him thoroughly competent to make such an investigation as you may desire.

Very truly yours, SELDEN E. MARVIN, *Sec'y.*

The above was written in reply to our letter addressed to Erastus Corning, Esq.

STEVENS INSTITUTE OF TECHNOLOGY,
DEPARTMENT OF ENGINEERING,
HOBOKEN, N. J., May 21, 1883.

My Dear Sir: I consider Mr. Porter one of the ablest, and I am sure he is one of the most scrupulously exact, men in the profession. I believe that you can trust him absolutely. His specialty is the steam engine * * * and if it is in his line, think no one could do it better.

Very truly yours, R. H. THURSTON.

SCHOOL OF MINES, COLUMBIA COLLEGE,
NEW YORK, May 23, 1883.

W. H. HODGMAN, Saratoga, N. Y.

Dear Sir: I take pleasure in replying at once to your inquiries in regard to Mr. Porter. He is one of the best mechanical engineers in this country. A man of large experience in steam-engines and machinery, and the inventor of the famous steam-engine which bears his name. I have known Mr. Porter for many years, and have never heard a word against him as a man. I should not know whom to recommend to you before Mr. Porter, of all the mechanical engineers now available.

Yours truly, W. P. TROWBRIDGE.

All of which is respectfully submitted.

DAVIS COLEMAN,
BENJ. W. CLAPP,
W. H. HODGMAN,
Committee.

FIRST REPORT.

DAVIS COLEMAN, ESQ., *President of the Board of Water Commissioners of the Village of Saratoga Springs.*

Sir: Having received from your Board, under date of May 25, 1883, instructions to make an examination of your new pumping machinery, which should be full and complete in every particular, I visited Saratoga on the 30th, for the purpose of a preliminary inspection, and on the first of June I made a careful internal examination of the pumps and engines.

The interval from the 12th to the 18th was employed in making preparations for a test, and in a detailed study of the machinery in operation, and on the last-named day a continuous trial of the boilers, engines, and pumps was commenced, and continued as follows:

A test for duty was first made in a run of sixty-eight hours, from 5 P. M. on the 18th to 1 P. M. on the 21st. The fire pressure was then put on, and the machinery was run under this pressure until 6 P. M. The pressure was then reduced to the usual domestic pressure, and the speed to that required to supply the demand for water.

On the morning of the 22d, at 8:30 o'clock, the engines were stopped for the first time during the trial, and one of them was disconnected, and the domestic service was then continued for two hours by one engine running alone, with which run the trial was concluded. I now present a preliminary report giving the results of the trial, and the conclusions at which I have arrived. A detailed report will be presented as soon as it can be prepared. I find as follows:

First. In the construction of the boilers, engines, and pumps, the requirements of the contract have been complied with in all respects, except that two Bulkley condensers have been supplied in place of the four air pumps and the jet condenser, which were specified. This change seems to have been a judicious one. The condensers maintain a good vacuum. The situation is favorable for condensers of this character, as the depth of the pit, in which is placed the hot well for the termination of their discharge pipes, is such that as soon as a partial vacuum is formed, the condensers will draw their water from the suction main without the aid of a pump, while the pressure in the delivery main is available to form the vacuum in starting. By this change four air-pumps have been dispensed with, improving the economy of running, and diminishing the liability to derangement.

Second. The design of the engines is, in one respect, a novel one. Each high-pressure cylinder is placed on the top of a low-pressure cylinder. The purpose and effect of this arrangement is to obtain a short and direct passage for the steam from the first to the second cylinder at each end.

With the exception of the beam engine designed by Mr. Leavitt, all previous plans of compound engines of this class have involved long passages between the high and low pressure cylinders, which are wasteful of steam. By the above happy expedient this objectionable feature is got rid of in these engines, and they are enabled to take rank among the most economical pumping engines. Special interest attaches to this trial, from the fact that these have been the first engines built after this design.

Third. In the details through which this general plan has been carried out, there are no features which seem open to criticism, but, on the contrary, all seem entitled to commendation. The construction is thoroughly mechanical in every respect. The forces are transmitted and the strains are resisted in the manner theoretically the most correct. The parts are strong and well proportioned, and the workmanship is excellent. The result is seen in the performance of the highest duty without any sign of labor. I do not think that the study of any machine has ever given to me a stronger feeling of confidence in its durability. A weakness had appeared, however in the sides of the pump chambers. In the original casting these sides were each stiffened by ten ribs two and a quarter inches high, but they were found to yield under the water pressure. This was remedied by two bolts carried through each chamber—eight in all. With respect to these bolts, it is to be observed that they resist the strains better than anything else would do; that if introduced originally they would properly have been regarded as an excellent feature, and that it is doubtful whether in other similar pumps anything so good could be substituted in place of them.

Fourth. As the pumps were working when I first saw them, and during my observation of them in domestic service, they delivered into the main for conduction to the village all the water that passed through them. There was no other open outlet for it.

The relief valve, the by-pass valves and the valves controlling the supply of water from the delivery main to the condensers were closed. When either of these was open even in the least degree, the rush of water through it was attended with a good deal of noise, and as the machinery moves in silence, except the sound made by the four steam valves in striking their seats, any such noise would be distinctly heard. I found every one of the six hundred and seventy-two valves in the pump chambers to be in its place and in good order, with the exception of a few of the suction valves in two of the chambers which were beyond my reach. The indicator diagrams taken from the four pump chambers show, however, conclusively, by the instantaneous and complete rise and fall of the pressure at the terminations of each stroke, that all the valves are in place and are perfect in their action.

No water returned through the old pumps. It could be lost in this way only by leakage past both delivery and corresponding suction valves. If either one were tight that would prevent it. These were examined and put in good order after my first visit, and afterward, on being run in my presence, the old pumps showed the same efficiency with the new ones, as nearly as they could be compared. Their perfect action was shown also by the absence of fluctuations of pressure in the delivery main. In compliance with the wishes of your Board, however, the valves which control the communication from these pumps to the delivery main were closed during the trial. It is, as I learn, the practice to keep these valves open, that all may be in readiness for starting the old pumps at any moment, and when so set open no loss of water takes place through them.

Fifth. As the water is delivered by the pumps direct into the distributing mains, its volume cannot be measured. The capacity of the pumps is found, therefore, by computation from the displacement of the plungers. The loss by return of water through the valves while closing, technically termed the slip, is very small, only a fraction of one per cent., and cannot be seen on the diagrams. I have, however, deducted one per cent. for slip. At the temperature of the water, seventy-one degrees, its expansion from its greatest density is only one-fourth of one per cent., and is disregarded.

There are two double-acting plungers, each twenty inches in diameter by thirty-six inches stroke, having rods four inches in diameter. Their average area is 307.88 square inches, and their aggregate travel is twelve feet at each revolution. Deducting one per cent. we have on each revolution 43,890 cubic inches, or just 190 gallons of water delivered. By computation made on this basis, the pumps at eighteen revolutions per minute delivered in twenty-four hours, **4,924,800 gallons**, or upward of twenty-three per cent. in excess of their required capacity.

Sixth. On the fire-pressure test, the machinery satisfactorily demonstrated its ability to run under a water pressure of one hundred and forty pounds on the square inch, at eighteen revolutions per minute, "with safety to all its parts." There was an entire absence of vibration or other indication of labor.

If the means had existed for getting rid of the water, I should have increased the speed to thirty revolutions per minute, as guaranteed in the contract. There can, however, be no question of the ability of the machinery to run at that speed, against one hundred and forty pounds water pressure, since all its parts then show a large factor of safety.

At that speed, it will deliver ninety-five gallons of water per second, which is more than twice its required capacity.

Seventh. The test for duty was made a long one, to insure that the duty shown should be no more than the machinery was capable of performing continuously. No allowance was made for friction anywhere. The duty is the net useful effect in water delivered. The pressures in the suction and delivery mains were reduced to the level of the center of the plungers.

Only two elements enter into this computation of duty, namely, the work done in delivering water, and the coal burned. The investigation of the performances of the boilers, engines, and pumps, which enabled the duty to be performed, will be made in my final report, but these have nothing to do with the computation of the duty.

About five o'clock on Thursday morning, sixty hours after the commencement of the run, a part of the packing blew out from the man-hole plate of one of the boilers. It was found impossible to stop this leak entirely, and I have therefore made the computation for sixty hours instead of sixty-eight hours.

During this time the engines made 68,255 revolutions, being an average of 18.96 revolutions per minute, and the average water pressure was eighty-seven pounds on the square inch.

Allowing one per cent. for the slip, there were done on each revolution 3,657.62 foot-pounds of work for each one pound of resistance on the square inch overcome in the main.

307.88 × 12 = 3,694.56, and deducting one per cent. leaves 3,657.62.

Multiplying this number by the revolutions and the pressure, as above, we have 21,720,100,000 foot-pounds of work done in the sixty hours in delivering water.

The coal was hand-picked and was of good quality, for it left only 4.73 per cent. of ashes and cinders. The whole quantity burned during this time, assuming the fires to have been in the same condition at the close as at the commencement, was 20,330 pounds, which shows a duty of **106,838,000 foot-pounds** for each one hundred pounds of coal consumed. The above assumption, however, was not fully warranted, since by omitting the first twelve hours the remaining forty-eight hours show, by a similar computation, a duty of only 102,340,000 foot-pounds for each one hundred pounds of coal consumed, and this may be taken as the real duty of the machinery in regular service. It exceeds the stipulated duty by 22,340,000 foot-pounds, or nearly twenty-eight per cent.

The first twenty hours show an apparent duty of **117,580,000 foot-pounds**, and the first twelve hours show an apparent duty of **127,170,000 foot-pounds.** The system of observation was such that the apparent duty at different periods of the test can be accurately ascertained, and the real distinguished from the fictitious.

Eighth. The fly-wheel on the engine is twelve feet and four inches in diameter, and its rim weighs 9,700 pounds. This is insufficient to maintain its motion when moving slowly, and so permits fluctuations of the water pressure, which increase in a rapid ratio as the motion of the engine is reduced.

It is not possible to impart uniform motion to a column of water by means of two double-acting plungers whose motions are controlled by cranks, as is the case here. The most favorable position for the two cranks is at right angles with each other, and so they are always set. The slowest motion of the water is when either crank is on one of its dead points. Midway between any two of these points, or when both cranks are at an angle of forty-five degrees, the motion is accelerated theoretically forty per cent. This is modified in these engines by the action of the connecting-rods, and the acceleration is made different in each of the four quadrants, being sixty per cent. in one and only twenty-five per cent. in another. This is the case when the rotation of the cranks is uniform, and is the best that can be done.

The fluctuations of pressure due to the above cause appear not to exceed five pounds. They could be reduced very much by employing a large air chamber, but I do not learn that this small amount of fluctuation is found very objectionable. The serious fluctuations are due entirely to the insufficiency of the fly-wheel, which at slow speed permits large variations in the motion of the cranks. The fly-wheel meets the requirements of the contract, but that does not make it sufficient.

A fly-wheel eighteen feet in diameter, having a rim of one hundred and thirty square inches in cross-section area, would have just five times the regulating power of the present one. With such a wheel all fluctuations above five pounds would disappear, even when running as slowly as four and one-half revolutions per minute. The present shaft would not carry this wheel, but the bearings and their supports are ample. A new shaft will need only to be larger between the journals, and its form on each side of the wheel would properly be a truncated cone.

The old pumps deliver the water without any appreciable fluctuation. This is because they consist of four double-acting pumps, driven by cranks, which are so set that their dead points divide the circle of revolution into eight equal parts, and these cranks are kept in very nearly uniform motion.

Ninth. With the above-described fly-wheel, one pump and engine, can, in case of necessity, be run alone entirely well. At present, with one engine, the fluctuations of pressure are excessive. There is also a liability for a single engine to be caught on the center. The fly-wheel should have holes at convenient distances apart, around its face, so that a bar can then be advantageously applied to move it off.

Tenth. The engines maintain with ease a fire pressure of one hundred and forty pounds on the square inch, and as already stated can deliver, under this pressure, ninety-five gallons of water per second. This pressure is the same that would be afforded by a reservoir in which the surface of the water was three hundred and twenty feet above the level of the pumping station. Now this would be a great head of water; but you have the very same thing, with the capacity for delivering a volume far greater than can be carried away.

Under these circumstances, to blame the pumping machinery, as I learn is being done, because only a small portion of this pressure can be realized in the hydrants in the village when streams are put on, is surely the very height of absurdity. The cause of this loss of pressure will have to be looked for where it exists. It must be obvious to every one, on reflection, that the engines can have nothing to do with it.

Very respectfully,

NEW YORK, June 28, 1883. CHARLES T. PORTER.

FINAL REPORT.

DAVIS COLEMAN, ESQ., *President of the Board of Water Commissioners of the Village of Saratoga Springs, N. Y.*

Sir: I have now the honor to present to your Board a detailed report of the trial of your new pumping machinery, supplementary to my general report already presented under date of June 28th, and to which reference is hereby made as constituting a part of this report.

The said machinery consists of two double-acting pumps, of a rated aggregate capacity of four million gallons per day, of two compound cylinder engines, each of which drives one of the said pumps, and of two boilers, which unitedly supply the engines with steam.

On the trial of this machinery, four separate continuous logs were kept, from 5 P. M. of the 18th to 8:30 A. M. of the 22d of June. These logs are here combined, and are presented in the following table, No. 1.

The observations were made simultaneously at the commencement of each hour an half hour, and diagrams from all the cylinders and pump chambers, making eigh engine and four pump diagrams, were taken directly afterward. Examples of each o these are given in connection with the description of the engines and pumps. The ob servations and diagrams were numbered consecutively, all those made or taken at th same time receiving the same number.

The heights of the barometer were furnished to me by the United States chief sig nal officer,—to whom my acknowledgments are due,—from the records of the Alban station, and are corrected for the elevation of the pumping station, two hundred an fifty-nine feet above tide-water. The thermometers used were supplied by C. J Tagliabue.

Eight Tabor indicators were employed, which, with two test gauges, were prepare by the Ashcroft Manufacturing Company expressly for this trial.

I had the assistance of eight young gentlemen from the graduating class of th School of Mines of Columbia College, who kindly volunteered their services, and t whose careful and intelligent work we are all indebted.

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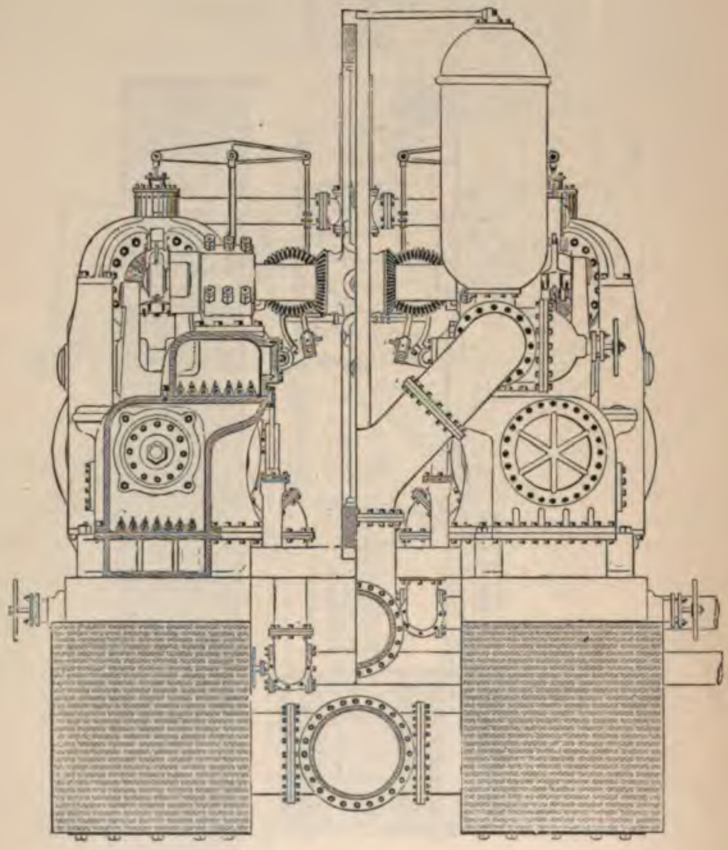


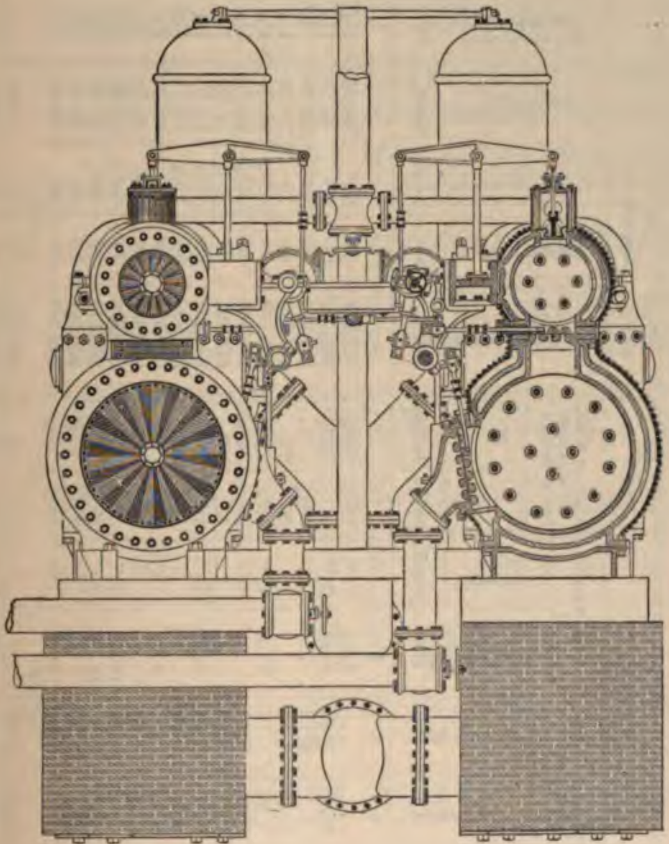


TABLE No. 1.

LOG OF EIGHTY-SEVEN HOUR TEST OF SARATOGA PUMPING MACHINERY, COMMENCED JUNE 18TH, 1883, 5 P. M.

Number of the Observation.	TIME.	BOILERS.								ENGINES.				PUMPS.			Temperature of the External Air.
		Weight of Coal brought in.		Time of Charging Fires.		Feed Water.		Height of Water above center line of Boilers.									
		Time.	Weight.	Boiler No. 1.	Boiler No. 2.	Time of Filling Tank.	Temperature.	No. 1.	No. 2.	Reading of Counter.	Steam Pressure.	Vacuum.	Height of Barometer.	Pressure in Supply Chamber.	Pressure in Delivery Main.	Fluctuation of Pressure in Delivery Main.	
			Lbs.				Deg. F.	Inches.	Inches		Lbs.	Inches.	Inches.	Lbs.	Lbs.	Lbs.	Deg. F.
	P. M.																
1	5.00	5.00	170	....	...	5.04	71°	19½	20½	933,273	68	25	29.85	1¼	....	..	60°
2	5.30	5.30	600	5.40	5.40	5.44	71°	....	19¼	933,752	84	26	29.83	1¼	95	5	60°
3	6.00	....	...	....	6.15	....	...	20¾	....	934,212	76	26	29.81	1¼	95	6	60°
4	6.30	....	...	6.30	6.55	....	...	20	20	934,676	83	26	29.79	1¼	90	5	61°
5	7.00	7.15	600	7.00	....	7.14	71°	19¾	19¾	935,118	79	25½	29.77	1¼	92½	5	62°
6	7.30	....	...	7.45	7.30	....	...	19½	19¼	935,568	79	25½	29.76	1¼	92	5	62°
7	8.00	....	...	....	8.00	....	...	19	19	936,031	78	25¾	29.75	1¼	93	5	62°
8	8.30	....	...	8.30	8.45	8.30	71°	18¼	18¾	936,497	80	25¾	29.75	1¼	112	5	62°
9	9.00	9.00	600	9.00	9.05	....	...	18	17¼	936,982	79	25½	29.74	1¼	102	4	62°
10	9.30	....	...	9.45	....	9.50	71°	16¼	17	937,511	79	25	29.73	1¼	99	4	61°
11	10.00	....	...	10.25	10.00	....	...	16	16¼	938,060	76	25½	29.72	1¼	96½	4	61°
12	10.30	....	...	....	10.45	10.38	71°	16	17¾	938,576	74	24½	29.72	1¼	97	4	61°
13	11.00	....	...	11.10	....	11.22	71°	17¼	17¾	939,119	75	24½	29.71	1¼	99	4	60°
14	11.30	11.30	600	11.50	11.30	....	...	17¼	17¾	939,661	75	25	29.70	1¼	99½	5	60°
15	12.00	....	...	....	0.10	0.07	71°	18¼	17¼	940,246	74	25	29.69	1¼	97	4	60°
	A. M.																
16	0.30	....	...	0.35	0.55	....	...	18	18	940,758	74	25	29.69	1¼	96	4	60°
17	1.00	....	...	1.20	....	1.14	71°	17¾	17¼	941,288	76	24½	29.68	1¼	98½	5	60°
18	1.30	1.30	600	...	1.30	....	...	16¾	16	941,797	74	25	29.67	1¼	98	6	60°
19	2.00	....	...	2.20	2 25	2.18	71¼°	15½	17	942,343	74	24½	29.66	1¼	99½	5	60°
20	2.30	...	...	....	....	....	...	16¼	15¾	942,870	74	24½	29.65	1¼	99½	5	59°

TABLE No. 1.—*Continued.*

LOG OF EIGHTY-SEVEN HOUR TEST OF SARATOGA PUMPING MACHINERY, CONTINUED JUNE 19TH, 1883, 3 A. M.

Number of the Observation.	TIME.	BOILERS.								ENGINES.				PUMPS.			Temperature of the External Air.
		Weight of Coal brought in.		Time of Charging Fires.		Feed Water.		Height of Water above center line of Boilers.									
		Time.	Weight.	Boiler No. 1.	Boiler No. 2.	Time of Filling Tank.	Temperature.	No. 1.	No. 2.	Reading of Counter.	Steam Pressure.	Vacuum.	Height of Barometer.	Pressure in Supply Chamber.	Pressure in Delivery Main.	Fluctuation of Pressure in Delivery Main.	
			Lbs.				Deg. F.	Inches.	Inches.		Lbs.	Inches.	Inches.	Lbs.	Lbs.	Lbs.	Deg. F.
	A. M.																
21	3.00	....	...	3.00	3.10	3.07	71°	15¼	17	943,498	72	24½	29.64	1¼	100	5	59°
22	3.30	3.50	600	3.40	3.50	3.55	70¾°	16½	15½	944,067	77	23½	29.64	1¼	103½	5	60°
23	4.00	....	...	....	....	....	...	15¾	16¼	944,670	76	24	29.63	1¼	98½	5	60°
24	4.30	....	...	4.35	4.40	4.45	70½°	16½	15¾	945,190	74	24½	29.62	1¼	98	6	60°
25	5.00	....	...	5.15	....	....	...	16	16¼	945,749	75	24	29.61	1¼	99½	5	60°
26	5.30	....	...	....	5.30	5.35	70½°	16¾	15¾	946,272	74	24½	29.60	1¼	99½	5	61°
27	6.00	6.00	600	6.00	6.10	6.21	70¼°	16½	16	946,876	72	25	29.60	1¼	97½	5	61°
28	6.30	....	...	6.40	6.40	....	...	16	16¾	947,439	74	24½	29.59	1¼	97½	5	61°
29	7.00	....	...	7.20	7.25	7.10	70°	16	17¾	947,993	76	24	29.58	1¼	98½	5	62°
30	7.30	7.40	600	7.55	....	....	...	17½	16	948,486	74	24½	29.58	1¼	99	6	62°
31	8.00	....	...	....	8.00	8.00	70°	17½	16½	949,067	70	24½	29.58	1¼	98	5½	62°
32	8.30	....	...	8.30	8.30	8.47	70½°	16½	18	949.706	76	25	29.57	1¼	99	4	62°
33	9.00	9.15	600	9.03	9.00	....	...	16	17½	950,358	76	24½	29.57	1¼	100	5	63°
34	9.30	....	...	9.23	9.40	9.32	72°	17	17½	950,897	75	24½	29.57	1¼	100	4½	63°
35	10.00	....	...	9.54	10.13	10.12	72½°	19	17	951,475	76	24½	29.57	1¼	99	3	64°
36	10.30	10.40	600	10.30	10.40	....	...	18½	18	952,029	76	24½	29.56	1¼	100	2½	72°
37	11.00	....	...	11.00	11.20	11.05	71°	18½	17½	952,592	80½	24¾	29.56	1¼	95	2½	73°
38	11.30	....	...	11.35	11.55	....	...	17	15½	953,129	80	24½	29.56	1¼	92	3	73°
39	12.00	12.15	600	12.05	....	12.12	71°	16½	16	953,682	79	24½	29.55	1¼	91	3	75°
	P. M.																
40	12.30	....	...	12.30	12.30	12.50	71°	16½	17	954,241	72½	24	29.55	1¼	83	2	74°

TABLE No. 1.—*Continued.*

LOG OF EIGHTY-SEVEN HOUR TEST OF SARATOGA PUMPING MACHINERY, CONTINUED JUNE 19TH, 1883, 1 P. M.

Number of the Observation.	Time.	Boilers. Weight of Coal brought in. Time.	Boilers. Weight of Coal brought in. Weight.	Boilers. Time of Charging Fires. Boiler No. 1.	Boilers. Time of Charging Fires. Boiler No. 2.	Boilers. Feed Water. Time of Filling Tank.	Boilers. Feed Water. Temperature.	Boilers. Height of Water above center line of Boilers. No. 1.	Boilers. Height of Water above center line of Boilers. No. 2.	Engines. Reading of Counter.	Engines. Steam Pressure.	Engines. Vacuum.	Engines. Height of Barometer.	Pumps. Pressure in Supply Chamber.	Pumps. Pressure in Delivery Main.	Pumps. Fluctuation of Pressure in Delivery Main.	Temperature of the External Air.
			Lbs.				Deg. F.	Inches.	Inches.		Lbs.	Inches.	Inches.	Lbs.	Lbs.	Lbs.	Deg. F.
	P. M.																
41	1.00	1.45	600	1.00	1.00	....	...	19½	17½	954,838	68	23½	29.54	1¼	79	2	76°
42	1.30	....	...	1.40	2.00	1.37	71°	18½	18	955,441	72	23½	29.53	1¼	82½	5	79°
43	2.00	....	...	2.20	2.30	....	...	17	17	956,068	70½	24	29.52	1¼	82	4	79°
44	2.30	....	...	2.55	2.45	2.50	71°	16½	16½	956,681	70½	24½	29.52	1¼	83	5	80°
45	3.00	3.20	600	3.25	3.10	....	...	18½	17½	957,245	72½	24	29.51	1¼	84	4	80°
46	3.30	....	...	....	3.35	3.30	71°	17½	19½	957,800	67½	24	29.51	1¼	78	4	79°
47	4.00	....	...	4.00	4.00	....	...	18¾	18½	958,360	75½	24	29.51	1¼	87½	5	79°
48	4.30	4.50	600	4.35	4.30	4.50	71°	17½	15½	958,929	77½	24	29.51	1¼	87	6	78°
49	5.00	....	...	4.45	4.45	....	...	15½	17½	959,512	70	23½	29.51	1¼	81	4	78°
50	5.30	....	...	5.20	5.00	5.37	71½°	16½	18½	960,033	69	24	29.50	1¼	81½	5	78°
51	6.00	....	...	6.00	5.15	....	...	17½	17	960,579	68	24	29.50	1¼	80	4	77°
52	6.30	6.40	600	6.30	6.00	6.50	71¾°	17	16	961,100	72	24	29.50	1¼	84	4	77°
53	7.00	....	...	6.50	6.40	....	...	16½	17½	961,653	68	24	29.50	1¼	81	5	75°
54	7.30	....	...	7.30	7.05	7.35	71½°	17½	17¾	962,181	68½	24	29.50	1¼	82	4	73°
55	8.00	....	...	7.50	8.00	....	...	18½	17½	962,665	65	24½	29.50	1¼	78½	5	71°
56	8.30	....	...	8.04	8.30	8.50	72°	17½	17	963,131	75	23¾	29.51	1¼	90	5	68°
57	9.00	9.15	600	8.45	9.07	....	...	17	17½	963,666	77	23¾	29.51	1¼	91	5	70°
58	9.30	....	...	9.30	9.15	....	...	16¼	16	964,137	74	24½	29.51	1¼	94	5	71°
59	10.00	....	...	10.15	10.00	10.05	72°	16½	16¼	964,723	74	24½	29.51	1¼	91	3	72°
60	10.30	....	...	10.53	10.40	....	...	15½	15½	965,255	77	24½	29.52	1¼	95	8	71°

TABLE No. 1.—*Continued.*

LOG OF EIGHTY-SEVEN HOUR TEST OF SARATOGA PUMPING MACHINERY, CONTINUED JUNE 19TH, 1883, 11 P. M.

Number of the Observation.	TIME.	BOILERS.								ENGINES.				PUMPS.			Temperature of the External Air.
		Weight of Coal brought in.		Time of Charging Fires.		Feed Water.		Height of Water above center line of Boilers.									
		Time.	Weight.	Boiler No. 1.	Boiler No. 2.	Time of Filling Tank.	Temperature.	No. 1.	No. 2.	Reading of Counter.	Steam Pressure.	Vacuum.	Height of Barometer.	Pressure in Supply Chamber.	Pressure in Delivery Main.	Fluctuation of Pressure in Delivery Main.	
			Lbs.				Deg. F.	Inches.	Inches.		Lbs.	Inches.	Inches.	Lbs.	Lbs.	Lbs.	Deg. F.
	P. M.																
61	11.00	11.00	600	11.08	11.19	11.18	72°	15¼	15	965,859	71	23¾	29.52	1¼	87	7	69°
62	11.30	....	...	11.44	11.59	....	...	15¼	15¾	966,495	72	23½	29.52	1¼	85	7	68°
63	12.00	....	...	....	....	12.00	72°	16½	15½	967,147	69	23½	29.52	1¼	83	5	68°
	A. M.																
64	0.30	0.45	600	0.30	0.45	0.47	72°	16½	14¾	967,770	70	23½	29.52	1¼	83	6	68°
65	1.00	....	...	1.11	....	....	...	16	17½	968,402	71	24	29.52	1⅛	82	4	69°
66	1.30	....	...	1.50	....	1.57	72°	17½	16½	968,989	74	24¼	29.52	1¼	86	5	69°
67	2.00	....	...	....	2.00	....	...	16½	15½	969,605	74	24¼	29.52	1¼	84	4	68°
68	2.30	2.40	600	2.45	2.48	2.45	72°	16	16½	970,187	77	24	29.52	1¼	86	5	69°
69	3.00	....	...	....	3.26	....	...	16¼	16¾	970,797	73	24	29.52	1¼	82	4	68°
70	3.30	....	...	3.35	....	....	...	16¼	15	971,369	75	24½	29.51	1¼	85	5	67°
71	4.00	....	...	4.13	4.20	4.05	72°	15	15½	971,984	79	24½	29.51	1¼	84	4	66°
72	4.30	4.35	600	4.37	4.53	4.50	72°	15	16	972,548	71	25	29.51	1¼	78	4	65°
73	5.00	....	...	....	....	....	...	16	16¼	973,116	76	25	29.51	1¼	82	5	65°
74	5.30	....	...	....	5.40	5.45	71½°	16¼	16½	973,663	74	25	29.51	1¼	82	5	66°
75	6.00	....	...	6.05	....	....	...	16½	16¾	974,239	73	25	29.51	1¼	81	6	66°
76	6.30	6.30	600	6.41	....	6.40	71°	16¼	16½	974,797	73	25	29.51	1¼	83	6	67°
77	7.00	....	...	7.20	7.20	....	...	17	17¾	975,377	73	24½	29.51	1¼	81	6	68°
78	7.30	....	...	7.58	7.59	7.30	72°	17¾	17¾	975,926	70	24½	29.51	1¼	78	7	69°
79	8.00	....	...	8.32	....	....	...	17½	16½	976,511	73	24½	29.51	1¼	82	5	71°
80	8.30	8.40	600	8.43	8.49	....	...	17	15	977,087	75	24½	29.51	1¼	83	6	72°

TABLE No. 1.—*Continued.*

LOG OF EIGHTY-SEVEN HOUR TEST OF SARATOGA PUMPING MACHINERY, CONTINUED JUNE 20TH, 1883, 9 A. M.

Number of the Observation.	TIME.	BOILERS.								ENGINES.				PUMPS.			Temperature of the External Air.
		Weight of Coal brought in.		Time of Charging Fires.		Feed Water.		Height of Water above center line of Boilers.		Reading of Counter.	Steam Pressure.	Vacuum.	Height of Barometer.	Pressure in Supply Chamber.	Pressure in Delivery Main.	Fluctuation of Pressure in Delivery Main.	
		Time.	Weight.	Boiler No. 1.	Boiler No. 2.	Time of Filling Tank.	Temperature.	No. 1.	No. 2.								
			Lbs.				Deg. F.	Inches.	Inches.		Lbs.	Inches.	Inches.	Lbs.	Lbs.	Lbs.	Deg. F.
	A. M.																
81	9.00	....	...	9.02	9.02	9.05	72°	15	16¼	977,690	73	24½	29.51	1¼	82	6	72°
82	9.30	9.50	600	9.30	9.13	9.55	72°	16¼	15¾	978,274	77	24	29.52	1¼	85	6	72°
83	10.00	....	...	10.00	9.40	....	...	16½	16¼	978,874	77	24½	29.52	1¼	85	6	74°
84	10.30	11.48	600	10.45	10.45	10.43	71°	16½	16¾	979,484	79	24½	29.52	1¼	87	6	75°
85	11.00	....	...	....	....	....	...	17¼	16¼	980,084	74	24½	29.52	1¼	82	5	75°
86	11.30	....	...	11.30	11.41	11.30	71°	17½	17	980,705	76	24½	29.52	1¼	83½	5	75°
87	12.00	....	...	12.15	11.55	12.20	71½°	17½	17¾	981,285	74	24½	29.52	1¼	81	4	78°
	P. M.																
88	12.30	....	...	....	12.35	....	...	17¾	18¼	981,874	74	24½	29.51	1¼	81	4	77°
89	1.00	....	...	1.20	....	1.05	71°	18	19	982,465	72	24½	29.51	1¼	80	4	79°
90	1.30	1.30	600	1.50	1.40	....	...	18¼	19¼	983,031	73	24½	29.50	1¼	80	4	79°
91	2.00	....	...	2.06	2.05	2.00	72°	18½	19	983,620	66	24	29.50	1¼	79	3	81°
92	2.30	2.57	600	2.50	....	2.55	71½°	19	18¾	984,215	70	24	29.49	1¼	82	4	80°
93	3.00	....	...	3.10	3.37	....	...	19½	18¾	984,794	68	24	29.49	1¼	80	4	81°
94	3.30	....	...	....	....	3.55	72°	19½	19	985,380	72	24	29.50	1¼	85	5	79°
95	4.00	....	...	4.37	4.20	....	...	18¾	20	986,006	74	24	29.50	1¼	86	4	78°
96	4.30	4.45	600	4.45	....	....	...	18½	19½	986,600	75	24½	29.51	1¼	87	6	75°
97	5.00	....	...	5.28	5.10	....	...	17¾	17¾	987,182	73	24½	29.52	1¼	84	4	73°
98	5.30	....	...	5.47	5.45	5.40	72°	17	15¾	987,775	69	24	29.53	1¼	80½	5	73°
99	6.00	6.12	600	6.26	6.06	....	...	17	16	988,381	77	24½	29.53	1¼	88½	7	71°
100	6.30	....	...	....	6.36	6.40	72°	16	15½	988,981	73	24	29.54	1¼	84½	5	71°

TABLE No. 1.—*Continued.*

LOG OF EIGHTY-SEVEN HOUR TEST OF SARATOGA PUMPING MACHINERY, CONTINUED JUNE 20TH, 1883, 7 P. M.

Number of the Observation.	TIME.	BOILERS.								ENGINES.				PUMPS.			Temperature of the External Air.
		Weight of Coal brought in.		Time of Charging Fires.		Feed Water.		Height of Water above center line of Boilers.		Reading of Counter.	Steam Pressure.	Vacuum.	Height of Barometer.	Pressure in Supply Chamber.	Pressure in Delivery Main.	Fluctuation of Pressure in Delivery Main.	
		Time.	Weight.	Boiler No. 1.	Boiler No. 2.	Time of Filling Tank.	Temperature.	No. 1.	No. 2.								
			Lbs.				Deg. F.	Inches.	Inches.		Lbs.	Inches.	Inches.	Lbs.	Lbs.	Lbs.	Deg. F.
	P. M.																
101	7.00	....	...	7.00	7.17	7.35	72°	16	15½	989,580	75	24½	29.55	1¼	87	6	69°
102	7.30	7.55	600	7.40	7.55	....	...	16½	15½	990,158	74	24	29.55	1¼	85	6	68°
103	8.00	....	...	8.18	8.10	....	...	15½	17½	990,769	72	24½	29.55	1¼	84	5	68°
104	8.30	....	...	8.55	8.35	8.20	72°	16½	15½	991,406	76	24¾	29.55	1¼	83	6	67°
105	9.00	....	...	....	9.07	....	...	16	16	991,992	79	24½	29.55	1¼	85	6	67°
106	9.30	9.40	600	9.38	9.20	9.50	72°	14¾	16½	992,568	77	24½	29.56	1¼	84	5	68°
107	10.00	....	...	....	10.05	....	...	16½	15	993,161	76	24½	29.56	1¼	84	4	67°
108	10.30	....	...	10.25	10.40	10.50	72°	15½	16	993,718	76	24½	29.56	1¼	85	6	66°
109	11.00	11.20	600	....	11.10	....	...	16	15½	994,298	77	24½	29.56	1¼	86	7	66°
110	11.30	....	...	11.52	11.25	11.50	72°	16	14½	994,877	74	24½	29.56	1¼	82½	5	66°
111	12.00	....	...	....	12.00	....	...	15½	15	995,457	76	24½	29.57	1¼	86½	4	66°
	A. M.																
112	0.30	....	...	0.25	0.35	0.50	72°	15	15	996,060	77	24½	29.57	1¼	86	3	66°
113	1.00	1.15	600	1.15	1.18	....	...	15½	15½	996,665	74	24½	29.57	1¼	83½	3	66°
114	1.30	....	...	1.45	1.30	1.50	72°	16	16	997,243	72	24½	29.58	1¼	82	3	66°
115	2.00	....	...	2.25	1.50	....	...	15½	16½	997,881	79	24½	29.58	1¼	87	3½	66°
116	2.30	2.42	600	3.00	2.30	2.30	72°	16¼	16½	998,520	70	24½	29.59	1¼	82	3	66°
117	3.00	....	...	3.25	2.42	....	...	16	15½	999,123	74	24½	29.59	1¼	83	4	64°
118	3.30	....	...	....	3.18	3.40	72°	15½	15½	999,714	74	24½	29.60	1¼	87	3	64°
119	4.00	....	...	....	4.00	....	...	15½	14¾	1,000,332	73	24½	29.60	1¼	85	3½	64°
120	4.30	4.30	600	4.38	4.30	4.45	72°	15½	14½	1,000,924	72	24	29.60	1¼	84	3½	64°

TABLE No. 1.—*Continued.*

LOG OF EIGHTY-SEVEN HOUR TEST OF SARATOGA PUMPING MACHINERY, CONTINUED JUNE 21ST, 1883, 5 A. M.

Number of the Observation.	TIME.	BOILERS.								ENGINES.				PUMPS.			Temperature of the External Air.
		Weight of Coal brought in.		Time of Charging Fires.		Feed Water.		Height of Water above center line of Boilers.		Reading of Counter.	Steam Pressure.	Vacuum.	Height of Barometer.	Pressure in Supply Chamber.	Pressure in Delivery Main.	Fluctuation of Pressure in Delivery Main.	
		Time.	Weight.	Boiler No. 1.	Boiler No. 2.	Time of Filling Tank.	Temperature.	No. 1.	No. 2.								
			Lbs.				Deg. F.	Inches.	Inches.		Lbs.	Inches.	Inches.	Lbs.	Lbs.	Lbs.	Deg. F.
	A. M																
*121	5.00	....	...	5.15	5.00	5.25	72°	15½	16	1,001,528	72	24½	29.60	1¼	84	3	64°
122	5.30	....	...	5.53	5.30	....	...	17	17	1,002,129	73	24½	29.61	1¼	85	5	62°
123	6.00	6.05	600	6.13	6.05	6.20	72°	17	16½	1,002,729	71	24	29.61	1¼	83	4	63°
124	6.30	....	...	6.38	6.48	....	...	16½	17	1,003,339	78	24	29.61	1¼	89	5	64°
125	7.00	7.25	600	7.13	7.20	7.25	72°	17	15½	1,003,950	76	24	29.61	1¼	86	4	64°
126	7.30	....	...	7.40	7.45	....	...	15½	16½	1,004,555	76	24½	29.61	1¼	83	3½	66°
127	8.00	....	...	8.10	8.18	....	...	16½	15¼	1,005,185	76	24½	29.62	1¼	85	5	67°
128	8.30	8.55	600	8.50	8.52	8.30	72°	16¼	16	1,005,799	76	24	29.62	1¼	83	3	68°
129	9.00	....	...	9.03	9.05	9.10	72°	16¼	16½	1,006,447	72	24	29.62	1¼	79	4	70°
130	9.30	....	...	9.35	9.37	9.55	72°	16½	18	1,007,075	76	23½	29.62	1¼	83	4	70°
131	10.00	10.27	600	10.14	10.12	....	...	17¼	17½	1,007,687	78	24	29.63	1¼	85	4	68°
132	10.30	....	...	10.32	10.38	10.50	72°	17¾	17	1,008,296	72	24	29.63	1¼	74	3	69°
133	11.00	....	...	11.17	11.33	....	...	17½	17¼	1,008,936	78	24	29.63	1¼	81	4½	70°
134	11.30	11.39	496	11.52	....	11.35	72°	16¼	17¼	1,009,580	73	24	29.63	1¼	82	4	71°
135	12 00	....	...	....	....	12.15	72°	17	18	1,010,207	77	24	29.62	1¼	82	4	72°
	P. M.																
136	12.30	....	...	....	12.32	12.50	71½°	17¼	18	1,010,869	74½	23½	29.62	1¼	93	4½	73°
137	1.00	1.00	600	1.03	1.05	....	...	18¾	20¼	1,011,532	70	23	29.62	1¼	105	3½	74°
138	1.30	1.35	600	1.30	1.28	....	...	17	18	1,012,049	78	25½	29.62	1¼	144	12	75°
139	2.00	....	...	1.55	2.05	2.00	71½°	17½	15½	1,012,519	83½	25	29.61	1¼	135	5	75°
140	2.30	2.50	600	2.11	2.34	2.30	71½°	13½	20	1,012,956	77	24½	29.61	1¼	140	5	75°

* Manhole packing of Boiler No. 2 blew out at 5 A. M. on June 21st.

TABLE No. 1.—*Continued.*

LOG OF EIGHTY-SEVEN HOUR TEST OF SARATOGA PUMPING MACHINERY, CONTINUED JUNE 21ST, 1883, 3 P. M.

Number of the Observation.	TIME.	BOILERS.								ENGINES.				PUMPS.			Temperature of the External Air.
		Weight of Coal brought in.		Time of Charging Fires.		Feed Water.		Height of Water above center line of Boilers.									
		Time.	Weight.	Boiler No. 1.	Boiler No. 2.	Time of Filling Tank.	Temperature.	No. 1.	No. 2.	Reading of Counter.	Steam Pressure.	Vacuum.	Height of Barometer.	Pressure in Supply Chamber.	Pressure in Delivery Main.	Fluctuation of Pressure in Delivery Main.	
			Lbs.				Deg. F.	Inches.	Inches.		Lbs.	Inches.	Inches.	Lbs.	Lbs.	Lbs.	Deg. F.
141	P. M. 3.00	....	...	2.45	3.13	3.00	71½°	18	16	1,013,411	74	25	29.61	1¼	140	5	76°
142	3.30	3.57	600	3.15	3.35	3.55	71½°	16½	18	1,013,820	69	25½	29.61	1¼	137	7	75°
143	4.00	....	...	3.40	4.20	....	...	18	17¾	1,014,248	68	25	29.62	1¼	137	6	75°
144	4.30	....	...	4.22	....	4.30	71½°	18	17½	1,014,668	66	25	29.62	1¼	137	7	75°
145	5.00	5.07	600	5.03	5.05	5.00	71°	17½	18½	1,015,067	65	26	29.63	1¼	139	12	73°
146	5.30	....	...	5.48	5.57	5.45	72°	18	18½	1,015,438	64	26	29.63	1¼	138	7	72°
147	6.00	....	...	....	....	....	...	18½	18½	1,015,824	62	26	29.63	1¼	137	7	72°
148	6.30	....	...	6.46	....	6.45	71½°	18½	19	1,016,088	62	23	29.64	1¼	86	12	70°
149	7.00	....	...	....	....	....	...	18½	19	1,016,331	60	25	29.64	1¼	90	10	69°
150	7.30	7.35	600	7.30	7.32	....	...	18¾	19	1,016,551	61	26	29.63	1¼	94	7	68°
151	8.00	....	...	8.00	....	8.05	72°	18¾	20½	1,016,779	66	26	29.62	1¼	98	10	68°
152	8.30	....	...	8.40	Shut off at 8.40 P. M.	....	...	18	Shut off at 8.40 P. M.	1,017,015	68	23	29.61	1¼	87½	9	66°
153	9.00	9.00	600	....		....	...	18½		1,017,180	63	26	29.60	1¼	86	12	66°
154	9.30	....	...	....		....	...	18½		1,017,344	61	25	29.59	1¼	88	10	64°
155	10.00	....	...	10.10		10.02	72°	18¾		1,017,524	59	25	29.58	1¼	80½	13	63°
156	10.30	....	...	....		....	...	18½		1,017,707	57	26½	29.57	1¼	92	10	63°
157	11.00	....	...	11.25		....	...	18½		1,017,872	58	25	29.56	1¼	94	10	62°
158	11.30	....	...	11.40		....	...	18		1,018,046	53½	26½	29.57	1¼	90	10	62°
159	12.00	....	...	12.00		....	...	18		1,018,197	49	26	29.58	1¼	92	10	62°
160	A. M. 0.30	....	...	0.20		0.30	72°	18		1,018,359	53	26	29.59	1¼	92	10	61°

TABLE No. 1.—*Continued.*

LOG OF EIGHTY-SEVEN HOUR TEST OF SARATOGA PUMPING MACHINERY, CONTINUED JUNE 21ST, 1883, 5 A. M.

Number of the Observation.	TIME.	BOILERS.								ENGINES.				PUMPS.			Temperature of the External Air.
		Weight of Coal brought in.		Time of Charging Fires.		Feed Water.		Height of Water above center line of Boilers.									
		Time.	Weight.	Boiler No. 1.	Boiler No. 2.	Time of Filling Tank.	Temperature.	No. 1.	No. 2.	Reading of Counter.	Steam Pressure.	Vacuum.	Height of Barometer.	Pressure in Supply Chamber.	Pressure in Delivery Main.	Fluctuation of Pressure in Delivery Main.	
			Lbs.				Deg. F.	Inches.	Inches.		Lbs.	Inches.	Inches.	Lbs.	Lbs.	Lbs.	Deg. F.
	A. M.																
*121	5.00	....	...	5.15	5.00	5.25	72°	15½	16	1,001,528	72	24½	29.60	1¼	84	3	64°
122	5.30	....	...	5.53	5.30	....	...	17	17	1,002,129	73	24½	29.61	1¼	85	5	62°
123	6.00	6.05	600	6.13	6.05	6.20	72°	17	16½	1,002,729	71	24	29.61	1¼	83	4	63°
124	6.30	....	...	6.38	6.48	....	...	16½	17	1,003,339	78	24	29.61	1¼	89	5	64°
125	7.00	7.25	600	7.13	7.20	7.25	72°	17	15½	1,003,950	76	24	29.61	1¼	86	4	64°
126	7.30	....	...	7.40	7.45	....	...	15½	16½	1,004,555	76	24½	29.61	1¼	83	3½	66°
127	8.00	....	...	8.10	8.18	....	...	16½	15¼	1,005,185	76	24½	29.62	1¼	85	5	67°
128	8.30	8.55	600	8.50	8.52	8.30	72°	16¼	16	1,005,799	76	24	29.62	1¼	83	3	68°
129	9.00	....	...	9.03	9.05	9.10	72°	16¼	16½	1,006,447	72	24	29.62	1¼	79	4	70°
130	9.30	....	...	9.35	9.37	9.55	72°	16½	18	1,007,075	76	23½	29.62	1¼	83	4	70°
131	10.00	10.27	600	10.14	10.12	....	...	17¼	17½	1,007,687	78	24	29.63	1¼	85	4	68°
132	10.30	....	...	10.32	10.38	10.50	72°	17¾	17	1,008,296	72	24	29.63	1¼	74	3	69°
133	11.00	....	...	11.17	11.33	....	...	17½	17¼	1,008,936	78	24	29.63	1¼	81	4½	70°
134	11.30	11.39	496	11.52	....	11.35	72°	16¼	17¼	1,009,580	73	24	29.63	1¼	82	4	71°
135	12 00	....	...	....	....	12.15	72°	17	18	1,010,207	77	24	29.62	1¼	82	4	72°
	P. M.																
136	12.30	....	...	....	12.32	12.50	71½°	17¼	18	1,010,869	74½	23½	29.62	1¼	93	4½	73°
137	1.00	1.00	600	1.03	1.05	....	...	18¾	20¼	1,011,532	70	23	29.62	1¼	105	3½	74°
138	1.30	1.35	600	1.30	1.28	....	...	17	18	1,012,049	78	25½	29.62	1¼	144	12	75°
139	2.00	....	...	1.55	2.05	2.00	71½°	17½	15½	1,012,519	83½	25	29.61	1¼	135	5	75°
140	2.30	2.50	600	2.11	2.34	2.30	71½°	13½	20	1,012,956	77	24½	29.61	1¼	140	5	75°

* Manhole packing of Boiler No. 2 blew out at 5 A. M. on June 21st.

TABLE No. 1.—*Continued.*

LOG OF EIGHTY-SEVEN HOUR TEST OF SARATOGA PUMPING MACHINERY, CONTINUED JUNE 21ST, 1883, 3 P. M.

Number of the Observation.	TIME.	BOILERS.								ENGINES.				PUMPS.			Temperature of the External Air.
		Weight of Coal brought in.		Time of Charging Fires.		Feed Water.		Height of Water above center line of Boilers.		Reading of Counter.	Steam Pressure.	Vacuum.	Height of Barometer.	Pressure in Supply Chamber.	Pressure in Delivery Main.	Fluctuation of Pressure in Delivery Main.	
		Time.	Weight.	Boiler No. 1.	Boiler No. 2.	Time of Filling Tank.	Temperature.	No. 1.	No. 2.								
			Lbs.				Deg. F.	Inches.	Inches.		Lbs.	Inches.	Inches.	Lbs.	Lbs.	Lbs.	Deg. F.
141	P. M. 3.00	....	...	2.45	3.13	3.00	71½°	18	16	1,013,411	74	25	29.61	1¼	140	5	76°
142	3.30	3.57	600	3.15	3.35	3.55	71½°	16½	18	1,013,820	69	25½	29.61	1¼	137	7	75°
143	4.00	....	...	3.40	4.20	....	...	18	17¾	1,014,248	68	25	29.62	1¼	137	6	75°
144	4.30	....	...	4.22	....	4.30	71½°	18	17½	1,014,668	66	25	29.62	1¼	137	7	75°
145	5.00	5.07	600	5.03	5.05	5.00	71°	17½	18½	1,015,067	65	26	29.63	1¼	139	12	73°
146	5.30	....	...	5.48	5.57	5.45	72°	18	18½	1,015,438	64	26	29.63	1¼	138	7	72°
147	6.00	....	...	....	....	....	...	18½	18½	1,015,824	62	26	29.63	1¼	137	7	72°
148	6.30	....	...	6.46	....	6.45	71½°	18½	19	1,016,088	62	23	29.64	1¼	86	12	70°
149	7.00	....	...	....	....	....	...	18½	19	1,016,331	60	25	29.64	1¼	90	10	69°
150	7.30	7.35	600	7.30	7.32	....	...	18¾	19	1,016,551	61	26	29.63	1¼	94	7	68°
151	8.00	....	...	8.00	....	8.05	72°	18¾	20½	1,016,779	66	26	29.62	1¼	98	10	68°
152	8.30	....	...	8.40	Shut off at 8.40 P. M.	....	...	18	Shut off at 8.40 P. M.	1,017,015	68	23	29.61	1¼	87½	9	66°
153	9.00	9.00	600	....		....	...	18½		1,017,180	63	26	29.60	1¼	86	12	66°
154	9.30	....	...	....		....	...	18½		1,017,344	61	25	29.59	1¼	88	10	64°
155	10.00	....	...	10.10		10.02	72°	18¾		1,017,524	59	25	29.58	1¼	80½	13	63°
156	10.30	....	...	....		....	...	18½		1,017,707	57	26½	29.57	1¼	92	10	63°
157	11.00	....	...	11.25		....	...	18½		1,017,872	58	25	29.56	1¼	94	10	62°
158	11.30	....	...	11.40		....	...	18		1,018,046	53½	26½	29.57	1¼	90	10	62°
159	12.00	....	...	12.00		....	...	18		1,018,197	49	26	29.58	1¼	92	10	62°
160	A. M. 0.30	....	...	0.20		0.30	72°	18		1,018,359	53	26	29.59	1¼	92	10	61°

TABLE No. 1.—*Continued.*

LOG OF EIGHTY-SEVEN HOUR TEST OF SARATOGA PUMPING MACHINERY, CONTINUED JUNE 22D, 1883, 1 A. M.

Number of the Observation.	TIME.	BOILERS.								ENGINES.				PUMPS.			Temperature of the External Air.
		Weight of Coal brought in.		Time of Charging Fires.		Feed Water.		Height of Water above center line of Boilers.		Reading of Counter.	Steam Pressure.	Vacuum.	Height of Barometer.	Pressure in Supply Chamber.	Pressure in Delivery Main.	Fluctuation of Pressure in Delivery Main.	
		Time.	Weight.	Boiler No. 1.	Boiler No. 2.	Time of Filling Tank.	Temperature.	No. 1.	No. 2.								
			Lbs.				Deg. F.	Inches.	Inches.		Lbs.	Inches.	Inches.	Lbs.	Lbs.	Lbs.	Deg. F.
	A. M.																
161	1.00	....	...	0.55	Not used.	....	...	18¼	Not used.	1,018,517	52	26	29.60	1¼	89	10	60°
162	1.30	....	...	1.15		....	...	18½		1,018,676	50	26	29.61	1¼	91	10	60°
163	2.00	....	...	2.05		2.25	72°	18¾		1,018,827	52½	26½	29.62	1¼	94	12	60°
164	2.30	....	...	....		....	...	18½		1,018,983	58	27	29.63	1¼	101	12	59°
165	3.00	3.10	600	3.00		....	...	17½		1,019,153	53	27	29.64	1¼	95	12	58°
166	3.30	....	...	....		....	...	18		1,019,307	59	26½	29.65	1¼	103	12	57°
167	4.00	....	...	4.00		....	...	18		1,019,473	52½	26	29.66	1¼	93	12	57°
168	4.30	....	...	4.50		4.30	72°	18½		1,019,629	54½	25¾	29.67	1¼	98	12	57°
169	5.00	....	...	5.50		....	...	19		1,019,805	52	27	29.68	1¼	91	10	57°
170	5.30	....	...	....		....	...	20¼		1,019,987	55½	26½	29.69	1¼	94	10	57°
171	6.00	....	...	....		6.00	71½°	21		1,020,157	51	26	29.70	1¼	86½	9	58°
172	6.30	....	...	6.45		....	...	20¼		1,020,357	54½	26	29.71	1¼	87	10	58°
173	7.00	....	...	7.05		....	...	19¾		1,020,566	58½	26	29.71	1¼	93	10	61°
174	7.30	....	...	7.55		....	...	19		1,020,772	54	25	29.71	1¼	84½	9	65°
175	8.00	....	...	....		8.00	72°	19		1,021.001	55	25	29.71	1¼	81½	7	65°

IN THE DESCRIPTION OF THIS SYSTEM OF PUMPING MACHINERY, AND PRESENTATION OF THE RESULTS OF THE TRIALS, THE NATURAL ORDER WILL BE TO FOLLOW THE COURSE OF THE POWER, COMMENCING WITH THE BOILERS, AND CONCLUDING WITH THE PUMPS.

FIRST.—THE BOILERS.

The dimensions and performance of the boilers are given in the following table, No. 2. The coal was brought into the boiler-house in a hand-car, in loads of six hundred pounds each, and was shoveled from the car directly into the furnaces. The furnaces were shortened by fire-bricks at the back, laid in cement.

It would not be practicable to empty the furnaces at the close of the test for duty, because the engines could not be stopped, and, moreover, it was necessary to proceed immediately to the test of the machinery under fire-pressure. I therefore took no account of the coal used in starting the fires, but commenced the observations when fresh and clean fires had been burning, and the engines had been running at their intended speed of about eighteen revolutions per minute, for nearly an hour, knowing that in a long trial the excessive apparent duty got during the early portion, from the combustion of coal that had not been weighed, could be readily ascertained and disregarded. The real rates of combustion and evaporation and engine performance, on the test for duty, are shown by the log during the forty-eight hours succeeding the first twelve hours, and this portion only will be considered in this connection. The water was measured in a tank, which was filled from the delivery main, and it was delivered to the boilers in quantities of fifty cubic feet. The weight of a cubic foot of the water was found to be sixty-two and a quarter pounds. It is certain that there could not have been an error in the measurement of the water so great as one per cent.

The natural draft of the furnaces is poor. This is to be accounted for, first, by the contracted air space in the grates; second, by the length of the horizontal flues relatively to the height of the chimney; and, third, by the location of the pumping station in a narrow valley.

A forced draft was employed, formed as follows: The ash-pit doors were tightly closed, and a cylindrical sheet-iron pipe, fourteen inches in diameter, was inserted into the side wall of the ash-pit under each boiler. These pipes turned upward outside the walls, and rose to the height of five feet. Their diameter at the top was sixteen inches, and they were contracted to a diameter of six inches in the middle of their vertical length. A steam jet was brought from the boiler in a three-quarter inch pipe, contracted at the end to half an inch, and terminating in an orifice one-eighth of an inch in diameter, located centrally in this air pipe, about a foot below the top. The force of the steam was regulated by a valve in the half-inch pipe. The induced air current was all that could be desired, and only a small portion of the full force of the jet was employed.

The performance of these boilers naturally attracts attention to this method of forcing the draft. It is obvious that the use of the steam jet must involve a loss, first of the heat contained in the steam used, and, second, from the dissociation of its gases, which latter only, at the most, could be made good by their complete recombination. The jet may, however, in addition to inducing a sufficient draft of air, help to a more intimate commingling of the oxygen and carbon in the combustion chamber, and so to a more perfect combustion.

It was not found practicable to take the temperature of the escaping gases in any satisfactory manner, but there can be no doubt that they were so much cooled, that in passing back over the top, being there further cooled by radiation, they rather caused a loss than a gain of heat in the boilers. The arrangement of the tubes was adapted to allow the best water circulation, and the evaporating surface and steam space were ample.

The evaporative performance of the boilers was in fact greater than the duty shown in the table, since the water condensed in the steam jackets was being continually returned to the boilers, to be evaporated again, and no account could be taken of its amount.

This will be found described in detail in the description of the engines. It is a matter of regret that the full evaporative duty of the boilers could not be ascertained

TABLE No. 2.

DIMENSIONS AND PERFORMANCE OF BOILER.

Type of boiler, cylindrical tubular.		
Number of boilers		2
Diameter "		66 inches.
Length "		18 feet.
Number of tubes in each boiler		87
External diameter of tubes		3 inches.
Arrangememt of tubes, horizontal and vertical rows.		
Least distance between tubes and shell of boiler		4.5 inches.
Height of crown of boiler aboves tubes		24 inches.
Heating surface in both boilers, as follows :		
One half the shells, 311 ; tubes, 2,460 ; total		2,771 sq. feet.
Furnace, width		6 feet.
" depth		5.75 feet.
" length of opening between grate bars		5 feet.
" " these openings uncovered		4.25 feet.
" width of each grate bar		1.75 inches.
" width of air spaces		.5 inches.
" square feet of grate area, under both boilers		51
" of which the proportion in air space is only		.22
Horizontal flues, as follows:		
Three times the length of each boiler	54 feet.	
From wall between boilers to chimney	24 feet.	
Total		78 feet.
Internal diameter of common flue and chimney		3 feet.
Height of chimney above flue		75 feet.
Square feet of heating surface for each one square foot of grate area		54.3
Square feet of heating surface for each indicated horse-power		14.9

PERFORMANCE OF BOILERS, ON THE TEST FOR DUTY, 48 HOURS, OBSERVATIONS 25 TO 121.

Coal burned	16,860 lbs.
" " per hour, on each one square foot of grate	6.9 lbs.
Water evaporated	155,961 lbs.
" " by combustion of one pound of coal	9.25 lbs.
Average temperature of feed water	71.5 deg.
" pressure of steam, by gauge	73.66 lbs.
Number of thermal units contained in one pound of steam	1,211.1
Equivalent evaporation from 212°, under one atmosphere	10.916 lbs.
Evaporation per hour from each one square foot of heating surface	1.175 lbs.

Performance on the Test under Fire Pressure, 5 hours, Observations 137 to 147.

Coal burned	2,800 lbs.
" " per hour, on each one square foot of grate	10.98 lbs.
Water evaporated	22,600
" " by combustion of one pound of coal	8.07 lbs.
Average pressure of steam, by gauge	72.06 lbs.
Evaporation per hour from each one square foot of heating surface	1.63 lbs.

It will be observed that the boilers were disproportionately large, and the high duty was obtained from them at a slow rate of combustion, and a very slow rate of evaporation.

The test under fire pressure was short; but the log shows unmistakably a falling off of twelve and a half per cent. in the evaporative efficiency of the boilers, when the greater pressure was put on the draft, required to produce nearly sixty per cent. more rapid combustion. This, however, is still moderate, and the evaporation per square foot of heating surface is still small. The performance of the boilers ought to have improved, instead of falling off.

I am inclined to attribute this loss of efficiency, as the force of the draft is increased, to the excessive width of the grate bars, which is three and a half times that of the air spaces between them. As the velocity of the air currents is increased, the combination with the carbon must from this cause become less complete. I recommend that the grate bars be changed. The coal was anthracite, supplied by the Delaware & Hudson Canal Co., and contained considerable slate; but, as will be seen from my first report, care was taken to get rid of all that. It burned freely, and made very little clinker.

SECOND.—THE ENGINES.

These are the horizontal compound-cylinder engines, of the type designed by Mr. H. F. Gaskill. The novel feature by which this type of engine is distinguished, is the location of the high-pressure cylinder on the top of the low-pressure cylinder, giving the short steam communication mentioned in my first report.

This arrangement requires the pistons in the two cylinders to move in opposite directions. These are connected with each other, and the pressure of steam upon them is transmitted to the plunger of the pump in the following manner: The latter is set in line with the low-pressure cylinder, and so receives the thrust produced by the pressure in that cylinder directly. The low-pressure piston is provided with two rods, which are extended to the length of ten feet to the cross-head by which they are connected to the rod of the pump plunger. Midway in their length, another cross-head is secured upon them, which runs on a guide. The rod from the high-pressure piston is keyed into a cross-head corresponding to this. The connection between these cross-heads is made by means of a short vertical beam and connecting links, through which also the force exerted in the high-pressure cylinder is transmitted.

A shaft carrying a fly-wheel revolves in bearings which are set upon the delivery chambers of the two pumps. The two engines are connected through this shaft by means of cranks set on either end of it, at right angles with each other, and connecting rods from these cranks to the upper ends of the beams. The cranks determine the length of the stroke, control the movements of the pistons and plungers, maintaining those of the two engines and pumps in a constant relation with each other, and give motion to the valve gear.

The steam is admitted to the high-pressure cylinders by double-seated poppet valves. A cut-off mechanism is introduced into the motion of these valves, which is adjustable to cut the steam off at any point in the length of the stroke. The point of cut-off may be fixed by hand, or by the action of the governor when the latter is thrown into gear. The governor is constructed to be operated by changes in the pressure of water in the delivery main, and acts to correct these changes by varying the point of cut-off, and thus the speed of the engine, in accordance with the variations in the demand for water.

The communicating valves between the high and low-pressure cylinders and the exhaust valves are gridiron slides.

All the cylinders and their heads are steam-jacketed. The water formed by condensation in the jackets is returned to the boilers in the following manner: The jackets blow directly into a small heater. The feed-water is conducted through this heater in a two-inch coil pipe. A separate pump receives its supply from the bottom of this heater, and sends the hot water forward to unite in the feed-pipe with the new water, which has been already partially warmed in passing through the coil in the heater. This is a restoration to the boilers of that portion of the heat supplied by them to the jackets, which had not been imparted from the jackets to the steam in the cylinders. The restoration is effected very well, except that the stuffing-box of the pump has no special provision for retaining hot water, and leaks sufficiently to make an appreciable loss.

There is no provision for preventing steam from blowing along with the water from the jackets into the heater. A valve was put into the drain-pipe, by partially closing which the full pressure was maintained in the jackets, and only a pressure of about twenty pounds was allowed in the heater. This was all that the situation admitted of, and it could not be known how much steam passed with the condensed water into the heater. Had it been practicable to attach an apparatus by which all the steam would have been kept back, and only the condensed water been allowed to pass, it would then have been easy, as it was desirable, by taking the temperature of the feed-water as it entered the boilers, to have known, very nearly at least, the quantity of steam condensed in the jackets. Since, however, the coil feed-water pipe in the heater acted as a surface condenser, and it was impossible to determine how much steam from the jackets was condensed by it, it seemed idle to take this temperature, which would rather lead to confusion than give any information.

The knowledge of the amount of heat imparted by the jackets would alone, however, give an exaggerated idea of the gain from their use, since there can be no doubt that they are most efficient in evaporating moisture contained in the steam at the time when it is passing directly into the condenser. The only satisfactory mode of determining the amount of the gain would be by a comparison of the results of this trial with those of another one, conducted with equal care, in which the steam was entirely shut off from the jackets. Such a trial is not suggested.

The following table, No. 3, gives the dimensions of the cylinders, and the next table, No. 4, shows the performance of the engines on the test for duty. Table No. 5 gives an analysis of the work done in the cylinders, and a comparison of this with the work done in delivering water, in eight examples. The observations selected, as representing a probable average, were those taken at the middle of each interval of six hours. It was found necessary to substitute No. 32 in place of No. 31, as at the time of taking the latter there was an exceptional fall of steam pressure for that part of the test.

TABLE No. 3.

CYLINDER DIMENSIONS.

Distance between centers of cylinders	45.5 inches.
Length of stroke in all cylinders	36 inches.
Number of high-pressure cylinders	2
Diameter " " "	21 inches.
" " " piston-rods	3 inches.
Area of each piston, average of both faces	342.83 sq. inches.
Piston displacement	12,342 cubic inches.
Waste room, in clearance and passages	494 cubic inches.
Proportion which this adds to piston displacement	.04
Addition to length swept through by piston	1.44 inches.
Total amount of displacement and waste room	7.428 cubic feet.
Number of low-pressure cylinders	2
Diameter " " "	42 inches.
Number of rods in each piston	2
Diameter " " "	3.5 inches.
Area of each piston, average of both faces	1,375.8 square inches.
Piston displacement	49,528.8 cubic inches.
Waste room in clearance and passages	1,347 cubic inches.
Proportion which this adds to piston displacement	.0272
Addition to length swept through by piston	.979 inches.
Total amount of displacement and waste room	29.442 cubic feet.
Proportion which is added by this waste room to the total capacity of high-pressure cylinder	.105

TABLE No. 4.

PERFORMANCE OF ENGINES ON FORTY-EIGHT HOUR TEST FOR DUTY.

Average number of revolutions made per minute	19.37
Average piston speed, in feet, " "	116.22
Pressure of the atmosphere, in pounds on the square inch	14.5
Average horse-powers exerted in high-pressure cylinders	109.2
" " " " " low " "	76.55
" " " " total	185.75
Pounds of coal burned per hour	351.
" " " " " horse-power per hour	1.9
Pounds of water evaporated per hour	3,250.
" " " " " horse-power per hour	17.5
" " " " " hour for each horse-power exerted in the high-pressure cylinders	30.
Total average back-pressure in high-pressure cylinders, pounds	12.5
Pounds of water per hour accounted for by indicator:	
1st. At point of cut-off in high-pressure cylinders	2,061.
Proportion of the quantity evaporated	.634
2d. At point of release in low-pressure cylinders	2,880.
Proportion of the quantity evaporated	.886
Temperature of the steam in all jackets, degrees	318.7
" " high-pressure exhaust (average)	204.
" " low " "	153.

TABLE No. 5.

PRESSURE OF STEAM IN THE CYLINDERS, AS SHOWN BY SELECTED INDICATOR DIAGRAMS, AND CORRESPONDING PRESSURE OF WATER, AND THE WORK DONE IN ENGINES AND PUMPS.

NUMBER OF THE DIAGRAM.	Initial steam-pressure above the absolute vacuum.	Point of cut-off.	Terminal steam-pressure above the absolute vacuum.	Mean effective pressure on the square inch.	Foot-pounds of work done in cylinders, per revolution.	Pressure of water.	Foot-pounds of work done in water delivered, per revolution.
	Lbs.	Inches.	Lbs.	Lbs.	Foot-pounds.	Lbs.	Foot-pounds.
No. 32.							
Right High-pressure.	87.70	11.52	36.50	50.08	103,013		
Left High-pressure	89.25	12.96	39.50	53.92	110,912		
Right Low-pressure. ..	26.75	. ...	9.13	9 59	79,147		
Left Low-pressure.	27.50	.....	9.00	9.56	78.949		
					372,021	97.75	357,765
No. 43.							
Right High-pressure.	82.25	9.32	29.75	43.14	88,738		
Left High-pressure	83.50	9.43	31.00	43.07	88.594		
Right Low-pressure........	22.50	....	7.38	7.57	62,505		
Left Low-pressure.........	21.75	....	7.50	6.94	57,255		
					297,092	80.75	295,545
No. 55.							
Right High-pressure.......	78.50	9.58	28 25	40.99	84,316		
Left High-pressure.	82.50	10.62	30.50	45 15	92,873		
Right Low-pressure.	21.87	.....	7.75	7.72	63,694		
Left Low pressure.........	22.87	.....	7.87	7.87	64,982		
					305,865	77.25	282,735
No. 67.							
Right High-pressure.	85.50	8.32	28.50	42.82	88,079		
Left High-pressure.........	84.75	9.47	29.00	44.90	92 359		
Right Low-pressure.......	22.75	....	7.87	7.58	62,604		
Left Low-pressure.....	21.00	....	7.50	7.20	59,435		
					302,477	82.75	302,865
No. 79.							
Right High-pressure... ...	84.00	7.88	26.50	40.64	83,596		
Left High-pressure........	84.25	9.29	28.25	43.71	89,911		
Right Low-pressure	21.00	....	7 25	6.50	53,689		
Left Low-pressure.........	20.75	...	7.25	7.16	59,137		
					286,333	80.75	295,545
No. 91.							
Right High-pressure.......	79.25	8 96	26.75	40.13	82,547		
Left High-pressure	79.75	9.97	28.50	42.46	87,340		
Right Low-pressure	22.00	....	7 25	6 78	55,968		
Left Low-pressure.........	21.50	....	7 00	7.39	61,019		
					286,874	77.75	284,565
No. 103.							
Right High-pressure.......	83.00	8.64	28.00	42.24	86,886		
Left High-pressure.........	84.25	9.90	29.50	44.90	92,359		
Right Low-pressure........	22.00	....	7.50	7.28	60,078		
Left Low-pressure..........	22.62	....	7 50	7.94	65,527		
					304,850	82.75	302,865
No. 115.							
Right High-pressure.......	86.75	9.18	29.75	45.06	92,687		
Left High-pressure.	88.50	10.01	32.25	48.58	99,928		
Right Low-pressure	23.87	.....	8.12	8.20	67,656		
Left Low-pressure.........	24.75	.....	8.00	8.86	73,154		
					333,425	85.75	313,845
Averages H. P...	83.98	9.69	30.16	44.90	311,117	83.19	304,466
L. P.........	22.84	....	7.74	7.76			

Average loss, per revolution, due to friction of engines and pumps, 6,651 foot-pounds; percentage of loss, 2.14 per cent.

Indicated horse-power, by above table, 182.6.

Corrected indicated horse-power (see text), 185.75.

Coal consumed in the forty-eight hours, 16 86 pounds.

Coal per indicated horse-power, per hour, 1.9 pounds.

I am disappointed in the result of these comparisons of the indicated power with the useful effect in the pumps. Irregularities appear, the causes of which, however, are obvious. The fluctuations of pressure in the main, rising and falling twice during each revolution, to different points and through different periods, rendered it difficult to fix the mean pressure with exactness. although much care was taken, and, on the whole, the errors doubtless balanced each other. The chief cause of the discrepancies was, however, that in the few minutes—generally about five—that necessarily elapsed between recording the observations and completing the taking of four pairs of cylinder diagrams, the pressure in the main was sure to change a little from variations in the demand for water, and the pressure in the cylinders would vary in like degree.

The average result shows a loss from all causes of .0214 of the indicated power. The loss in the passage of water through the valves was .016 of the pressure, which would leave only about one-half of one per cent. for friction of the machinery. This friction was small, but the power lost in it was a good deal more than that.

The average water pressure in these selected cases was 83.19 lbs. The real average during the forty-eight hours test for duty was 84.57 lbs. I have assumed this error to afford a measure of a corresponding error in the indicated power exerted, and have accordingly increased this from 182.6 horse-powers, the measure of the diagrams, to 185.75 horse-powers, which I assume to be the average power really exerted during the test for duty.

The four pairs of engine diagrams, No. 103, are given as fair examples. Two theoretical curves are drawn on them. Of these, the upper one is the isothermal curve, representing the Mariotte law of expansion, which, on account of the readiness with which the height of its ordinates can be determined, is employed as the standard reference curve for indicator diagrams. The lower curve is the curve showing equal weights of steam at the different densities. This falls about midway between the isothermal and an imaginary curve, not shown here, which is known as the adiabatic curve, or curve of no transmission, and which represents the expansion of steam in a cylinder formed of a non-conducting material, impelling a similar piston that moves without friction, so that the steam neither receives nor imparts heat, but only loses that which is converted into work. This curve used to represent the pressure varying as the reciprocal of the seventeenth power of the sixteenth root of the space occupied by the steam, but this is now changed to the tenth power of the ninth root, nearly.

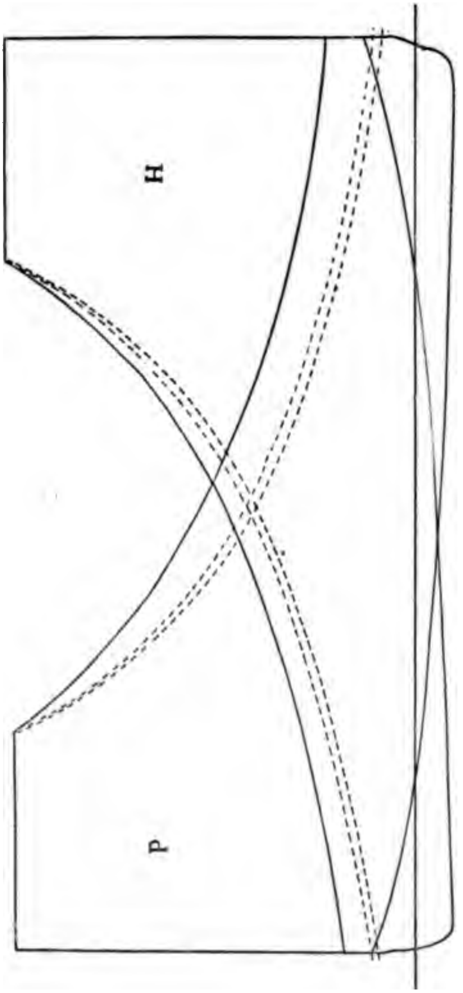
One of these high-pressure diagrams, and the corresponding low-pressure diagram, designated by the letters L P, have been combined to form the diagram prefixed to this report [pp. 30–31], which represents the expansion of the steam through the two cylinders.

The facts of practical consequence learned from these diagrams, in addition to the demonstration which they afford of the excellent working of the valve gear of these engines, are the following, as obtained from the examination of all the diagrams. These facts are not at all peculiar to these engines, but they would be equally true of all engines working under similar conditions, namely, at the same piston speed, equally well steam-jacketed, with the same pressure and back-pressure, and cutting off at the same point in a stroke of the same length :

First. A little more than one-third of all the steam evaporated in the boilers is condensed as it enters the high-pressure cylinders, and exists in the state of water at the point of cut-off. The exact proportion is .366 of the whole. This computation is not affected by the steam condensed in the jackets. The water thus formed, being in these engines returned directly to the boilers, may be considered as a separate body of water continually circulating in this manner, and so not affecting the quantity supplied to the boilers, but only bringing to the cylinders continually an additional supply of heat.

COPIES OF DIAGRAMS, NO. 103.

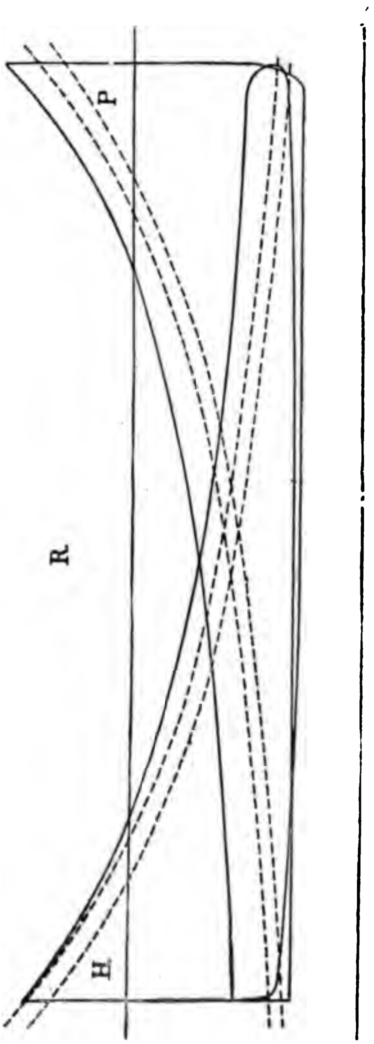
RIGHT HIGH-PRESSURE. Scale, $\frac{1}{35}$.



Second. During the expansion in the high-pressure cylinders, in addition to th reevaporation of water formed by the conversion of heat into mechanical work, abo one-half of the above-mentioned water was reëvaporated, increasing very much tl pressure at the end of the stroke.

Third. All the heat contained in the steam before it entered these cylinders, with that was imparted to it from the jackets while there, except only the small porti converted into work, passed into the low-pressure cylinders, and thence it, togeth with that received from the low-pressure cylinder jackets, all except that which w there converted into work, passed into the condensers.

Right Low-Pressure. Scale, $\frac{1}{12}$.

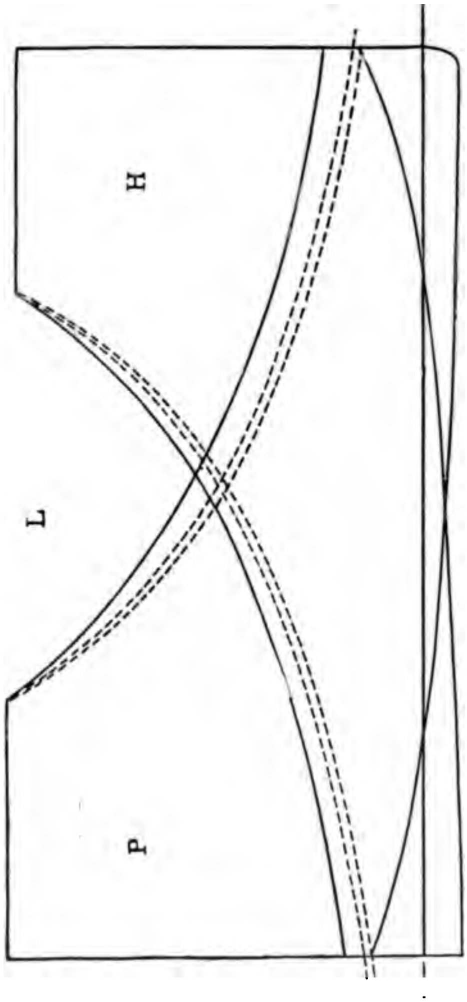


th. When the communications to the low-pressure cylinders were opened, a fall sure in the high-pressure cylinders to the amount of ten per cent. was ıtely necessary, in order to fill the waste room in those cylinders; all fall of pres- yond that amount, at that point, was caused by condensation of the steam in ; those cylinders.

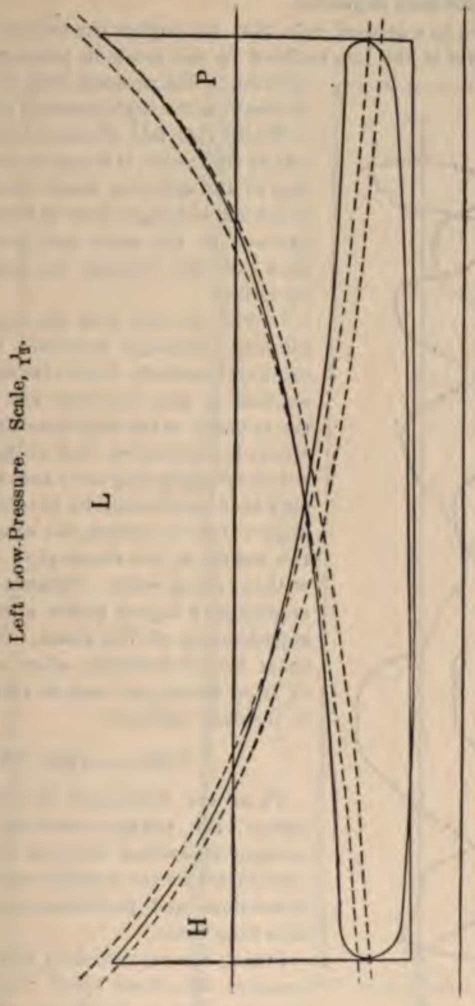
ı. This loss is afterward made good by reëvaporation in the low-pressure cylin- d something more, so that at the end of the stroke 88.6 per cent. of the water l to the boilers is present in the state of steam, and only 11.4 per cent. is in the water.

COPIES OF DIAGRAMS, NO. 103.

Left High-Pressure. Scale, $\frac{1}{40}$.



Sixth. The most important fact disclosed is, that the earlier the cut-off in the hi pressure cylinders, and so the lower the point reached by the expansion, the less v be the proportion of the steam condensed in entering the low-pressure cylinders. seems that the jackets can maintain these at only a moderate temperature, and, th fore, the higher the pressure at which the steam is exhausted into them the greater quantity that is instantly condensed. There was some want of uniformity in results, and a difference in this respect between the two engines that I cannot acco for, as I made sure that the valves were all tight; but the above fact very cle appears. It was well illustrated during the fire test, when the steam was allowe follow the pistons nearly to the end of the stroke. Thus, in diagrams taken from right-hand engine, No. 90, the fall at one end of the cylinder, from a terminal pres



Left Low-Pressure. Scale, $\frac{1}{15}$.

urteen pounds above the atmosphere, was four and a quarter pounds, and at the r end, from a terminal pressure of twelve pounds, the fall was four pounds, showing a pound and a half lost by condensation; while diagrams No. 141 show at one end, a terminal pressure of fifty-two pounds above the atmosphere, a fall of fifteen ds, and at the other end, from a terminal pressure of forty-seven pounds, a fall of nteen pounds.

he left-hand engine, in diagrams No. 40, shows at one end, with a terminal pressure fteen pounds, a fall of seven pounds, and at the other end, with a terminal pressure ighteen pounds, a fall of eight pounds; but in diagrams No. 141, with a terminal sure of fifty pounds at one end, it shows a fall of twelve pounds, and at the other end, with a terminal pressure of fifty-one pounds, it shows a fall of twenty-three pounds. These diagrams have not been engraved.

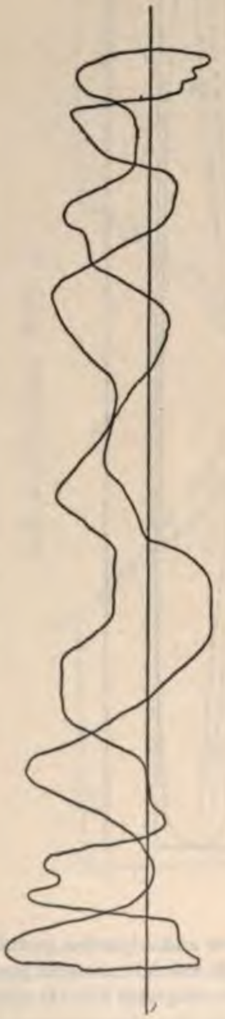


DIAGRAM No. 6.

Showing Pressures in the Supply Chambers of Pumps. Scale, $\frac{1}{12}$.

It may be laid down as a general rule that the higher the terminal pressure in the first cylinder the greater is the loss suffered by the steam in passing into the second cylinder. The pressure falls because the supply of steam in the high-pressure cylinder is limited.

While this fall affords a measure of the heat lost at that point, it suggests what the condensation of the entering steam must be in cylinders in which, although their surfaces have just been exposed to the same cool vapor, the pressure does not fall, because the supply of steam is unlimited.

It will be seen that the degree of economy attained, although excellent when judged by existing standards, is still far short of that which engines of this character are capable of; that the economy in the high-pressure cylinders, taken alone, is inferior to that obtained in first-class non-condensing engines: and that further economy is to be obtained by preventing, in a greater degree than at present, the transfer of heat from the steam in the steam-pipe to the condenser without doing work. This can be done either by employing a higher piston speed or by moderate superheating of the steam. A combination of these would, doubtless, effect a further increase of from twenty per cent. to twenty-five per cent. in the duty obtained.

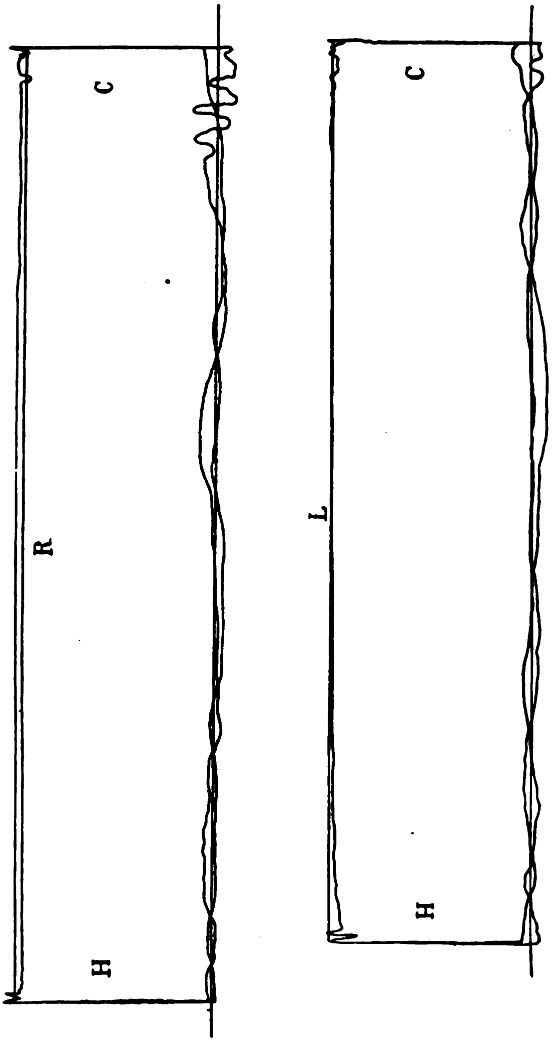
THIRD.—THE PUMPS.

These are horizontal double-acting plunger pumps, the plunger working through a diaphragm separating the two chambers, and are constructed on the multiple-valve system. Their dimensions and performance are given in the following table.

During the test for duty and a part of the fire pressure test, their speed very much exceeded that required to supply the demand for water.

The excess of water delivered was got rid of partly by opening hydrants and partly through the relief valve.

The pressure in the supply chambers was got by the indicator, with a spring showing twelve pounds to the inch. The pencil described the figure shown in diagram No. 6, which affords an interesting exhibition of the changes in the pressure as momentum is imparted or arrested in the current of water. The mean pressure, when corrected for elevation, was found to be one and one-quarter pounds on the square inch at the center of the plungers, showing the level of the water from which the supply is drawn to be about thirty-four inches above this point.

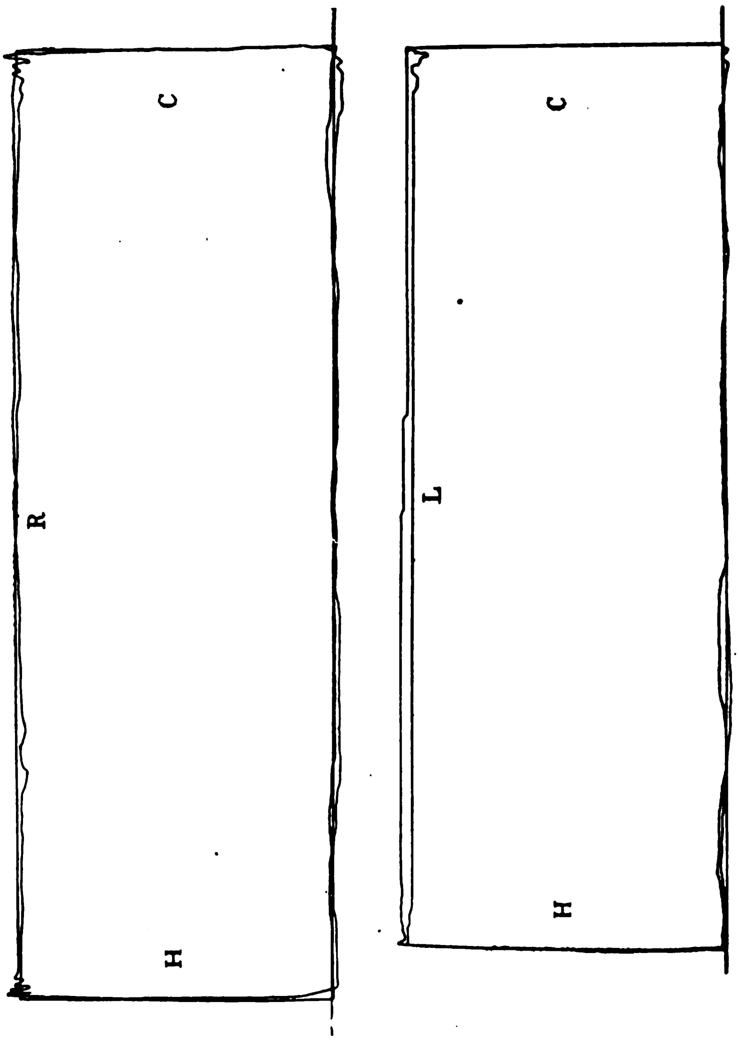


DIAGRAMS FROM THE PUMPS.

Scale, $\frac{1}{10}$.

No. 97.

t is to be observed, relatively to the fluctuations of pressure, both in the supply nber and in the delivery main, and also relatively to the motion of the plungers le the valves are closing, that in these pumps this motion, with that of the pistons, ontrolled by the cranks, precisely as if it was transmitted through them. The 'es are assumed to close only as fast as gravity would cause them to fall, although ıout doubt the reverse current brings them to their seats more quickly than this. even on this assumption, the slip is no more than four-tenths of one per cent.



No. 144.

The pumps are provided with by-pass pipes and valves, forming, when the latter are open, a communication between the two chambers of each pump. Their function is this: At the instant of starting, the engines are able to exert only a portion of their power, because no steam has yet been admitted to the low-pressure cylinders. The pressure of water is liable to be greater than the steam in the high-pressure cylinders alone can overcome.

In such a case the by-pass valves can be opened, when the water flows through them into the opposite chambers, the pressure on the opposite ends of the plungers is in a measure equalized, and the engine is enabled to start. As soon as it is fairly in motion, these valves are closed.

They are available, also, when the motion of the engines required to supply the demand for water would be so slow that, on account of the insufficiency of the fly-wheel, it could not be maintained with the desired regularity, or perhaps not at all. Then, by partially opening these valves, the engines are enabled to run faster, and so to maintain their motion satisfactorily, the excess of water displaced by the plungers passing through them into the chambers that are being filled.

This is a somewhat wasteful use of steam, but otherwise the expedient is not objectionable.

The diagrams are exact reproductions of the originals.

To these diagrams from the pumps an addition must be made of one pound in each forty pounds, as the indicator springs are corrected for the temperature of 212°, and at the temperature of the water are too strong in the above degree.

TABLE No. 6.

DIMENSIONS AND PERFORMANCE OF PUMPS.

Number of pumps	2
Aggregate length of the two chambers in each pump	7 feet 6 inches.
Width " " " " "	2 feet 3 inches.
Height " " " " "	2 feet 11 inches.
Capacity of each pump chamber, in cubic feet	25.
Diameter of plunger, inches	20.
" " rod	4.
Area of plunger, mean of two faces, square inches	307.88
Stroke of plunger, inches	36.
Displacement of plunger, in cubic feet	6.41417
" " " in gallons	48.
Number of double strokes of each plunger per minute	19.37
Mean velocity of plunger, in feet, per minute	116.22
Number of valves acting together	84.
Diameter of each valve-opening, inches	1.3125
Total area opened by valves, square inches	113.6
Lift of each valve, inches	.3125
Mean velocity of water through valves, in feet, per second	5.25
Weight of each valve, pounds	.5625
" " " " for each one square inch of opening, pounds	.416
Resistance to current on each one square inch of opening, pounds	.26
Total loss of efficiency in pounds on square inch	(.416+.26)×2=1.352
Mean excess of pressure in the delivery main, pounds on square inch	84.57
Proportion of power lost in passage through valves	.016
Time occupied by valves in closing, seconds	.04
Motion of plunger while valves are closing, inches	.07
Proportion of stroke lost in closing of both admission and delivery valves	.004
Net delivery from both pumps per double stroke, gallons	190.
" " " " " " " pounds	1,584.

Delivery per day at eighteen revolutions per minute, gallons.	4,924,800.
Number of double strokes made in 48-hour test	55,779.
Net foot-pounds of work done on each double stroke	309,338.
" " " " " " by consumption of 100 lbs. of coal	102,340,000.

FOURTH.—TEST OF PUMPS UNDER FIRE PRESSURE.

This test, so far as the pumping machinery is concerned, was of an entirely satisfactory character. The log, from observation No. 137 to observation No. 147 inclusive, shows the performance during this test. At the instant of one o'clock, as will be seen, the water pressure had only begun to rise, but it reached one hundred and forty pounds in a very few minutes.

I have given my conclusions from this test fully in my first report, and have nothing to add to them.

One of my assistants accompanied Dr. Hodgman, of your Board, and under his directions applied a pressure gauge to a number of hydrants while the fire pressure was maintained at the station. The pressures shown are exhibited in the following table, No. 7. It is, of course, understood that pressure at the hydrant is the only thing that determines the force of a stream of water through any given hose and nozzle.

On the previous day, I applied the gauge to the hydrant at the pumping station, and found the pressure in it to be the same as shown in the pumps, with the allowance for the difference of level.

TABLE No. 7.

PRESSURES OF WATER MAINTAINED IN THE FIRE HYDRANTS WITH A PRESSURE OF 138.5 LBS. IN THE PUMPS.

LOCATION OF HYDRANT.	Elevated above Tide Level.	Static Pressure Obtained in Hydrant.	Static pressure, Corrected to Level of Pumping Station.
Loughbury Water Works	259 ft.	138.5 lbs.	138.5 lbs.
Walton and Broadway—See Note 1	295 "	73 "	88.5 "
Caroline and Broadway—See Note 2	290 "	80 "	93 "
Division and Broadway—See Note 3	292 "	88 "	102 "
Washington and Broadway—See Note 3	286 "	90 "	101 "
West Congress and Broadway—See Note 4	273 "	100 "	106 "
Franklin and Washington " " "	299 "	96 "	113 "
Ash and South Franklin " " "	301 "	91 "	109 "
Circular and Union Avenue " " "	291 "	92 "	106 "
George and Nelson " " "	299 "	95 "	112 "
Caroline and Regent " " "	295 "	98 "	113 "

NOTE 1.—While this pressure was taken, there were six streams open on Broadway below and none above.

NOTE 2.—At this time, there were three streams above and none below.

NOTE 3.—At these times, there was one stream below and none above.

NOTE 4.—At these times, there were no streams on.

Correction for one foot of elevation, 0.43 lbs.

FIFTH—THE DOMESTIC SERVICE.

At the close of the fire pressure test, the speed and pressure were reduced to those required for the domestic service, and a record of the performance of the machinery was kept for fourteen hours. (Observations 148 to 175.) During this time the speed varied, according to the demand for water, from five revolutions to nearly eight revolutions per minute. One wonders where so much water is used during the night.

Respecting the fluctuations of pressure, it will be observed that on the test for duty these rarely exceeded five pounds, and sometimes fell to three pounds, and two and one-half pounds, and one to two pounds; but that during the fire pressure test they ranged higher, although the speed was about the same, and that on the domestic service, when the speed was reduced, the fluctuations were from ten to twelve pounds. The cause of this increased fluctuation was that the steam was all this time permitted to follow the piston nearly to the end of the stroke. The uniformity of rotative effect from the steam pressure is most nearly approximated in double engines when the steam is cut-off at about the half-stroke, or earlier. The improved regularity of the pressure in the main during the test for duty was produced by cutting off the steam, as shown in the diagrams. I had proved this before. On my first visit to the engines, the cut-off was not in gear, and the fluctuations were from ten to twelve pounds. At my suggestion, the engineer set the valve gear to cut-off at about one-half stroke, when the fluctuations fell to about five pounds, and did not exceed this in the day-time afterward. Regularity as well as economy will be promoted by working the steam expansively in the high-pressure cylinders. A good degree of regularity is not, however, at present attainable by any means when the engines are moving very slowly.

SIXTH.—THE USE OF A SINGLE ENGINE.

The experiments with the engines were concluded by disconnecting one engine, by removing its connecting-rod, and running the other engine and pump alone for two hours. Although this ran at double speed, still it halted on each center badly, and the fluctuations of pressure were from twelve to fifteen pounds. I should judge that in the night, or during the winter, when the motion is slow, with the present light fly-wheel there would be a liability to stoppage on the centers, should it ever become necessary to use one engine alone. While being run in this way it stopped, twice, on being shut down, just before reaching the center, and then it appeared that provision had not been made for this emergency, and considerable delay was suffered while several men were exerting themselves to get it over. The changes suggested in my first report will remedy these defects, which are at present a blemish on an engine so excellent in every other respect that I regret exceedingly having thus to qualify what otherwise must be entire commendation. Very respectfully,

New York, July 28, 1883. CHAS. T. PORTER.

Record of Average Duty of the First Gaskill Pumping Engine at Saratoga Springs, N. Y., on Regular Daily Service, as Kept by the Engineers in Charge, David L. Holland and George F. Bacon.

Average duty for the year 1884	105,425,937
“ “ “ “ 1885	101,700,644
“ “ “ “ 1886	105 940,690
“ “ “ “ 1887	106,318,872
“ “ “ “ 1888	108,401,880
“ “ “ “ 1889	107,676,411
Average duty for the six years, figured direct from total coal consumed, with no deductions whatever for ashes, steam for heating, or other purposes	105,910,739

COLUMBUS, OHIO.

FEBRUARY, 1884.

REPORT OF PROF. T. C. MENDENHALL,

PRESIDENT OHIO STATE UNIVERSITY.

(SUPERINTENDENT COAST AND GEODETIC SURVEY, WASHINGTON, D. C., 1889.)

OHIO STATE UNIVERSITY,
COLUMBUS, O., April 10, 1884.

To the Trustees of the Water Works:

GENTLEMEN—Concerning the tests of the new Gaskill Pumping Engine, which was recently put in place at the Water Works in this city, I have the honor to report as follows:

By your contract with the Holly Manufacturing Co., this engine was guaranteed to come up to a specific standard in the performance of its work before being finally accepted. This test was for the purpose of determining whether or not it was in accordance with the guarantee, and I was invited by you to be present, and to take such notes and make such observations as would enable me to calculate the duty of the engine, according to the conditions specified in the contract. With your permission I invited S. W. Robinson, Professor of Mechanical Engineering, and F. H. Eldridge, Professor of Steam Engineering of the Ohio State University, to be present and take charge of such parts of the tests as they might select. Professor Robinson was only able to be present a part of the time, but Professor Eldridge was present the entire day, and had direct supervision of such observations as were made in the boiler room. In addition, eight or ten students in the University volunteered their services and rendered invaluable assistance in making the continuous series of records.

The regular tests were begun at 10 A. M., on February 23, 1884. At the end of each period of two hours the water and coal accounts were carefully "squared up," so that each period of that length may be considered as complete in itself. This was done in order that any interruption by an alarm of fire might not destroy the value of more than two hours' work. Three such periods were regularly completed by 4 P. M. The fourth, which was only one and a half hours long, was disturbed by a fire alarm, during which the pressure was maintained for a short time at about 120 pounds per square inch. The regular readings and weighings were kept up, however, and the results are given with those of previous hours in the accompanying tables. The response to the alarm was very prompt, the high pressure having been reached in less than one minute after the stroke of the gong. Although the pressure fluctuated somewhat both at the beginning and end of the high pressure period, the whole disturbance was so short that it is not believed that the integrity of the results was materially interferred with.

Although not involved in the regular tests, the question of the uniformity of motion of the fly-wheel during a single revolution was examined. For this purpose, one of the shafts communicating motion to the valves was made use of, as through gearing with the main shaft, and it reproduced the motion of the fly-wheel very exactly. The diameter of this shaft was about four inches. About a foot of its length was covered with paper, which was smoked over its entire surface. A tuning fork making about one hundred vibrations per second was arranged so that a light style on the extremity of one of its prongs would impinge upon the surface of the smoked paper, and at the same time the fork could be given a fairly uniform motion in the direction of the axis of the shaft. In this way the vibrations of the fork were written upon the paper during several revolutions of the shaft. The paper being then removed, the variations in the angular velocity of the shaft could be easily examined. This was done, of course, by counting the number of vibrations written upon a given linear space around different parts of the shaft. Several sheets were prepared in this way, and an examination of them revealed a slight periodic variation in speed, the amount of which depended on the speed of the engine. At the rate of running during the most of the time of the tests, the deviation from regularity was very small.

The following table will exhibit the observations made during the tests with computations of duty based upon them. For the arrangement and reduction of these observations, I am indebted to Mr. E. H. Mark, who was present during the tests and assisted in making and recording observations.

NUMBER OF OBSERVATIONS.	Time.	BOILER No. 1.				BOILER No. 2.				BOILER No. 5.				WATER.				No. of lbs. of water evaporated for 100 lbs. coal consumed.
		No. of lbs. of coal weighed in.	No. of lbs. of ashes weighed out.	Total number of lbs consumed.	Readings of steam gauge.	No. of lbs. of coal weighed in.	No. of lbs of ashes weighed out.	Total number of lbs. consumed.	Readings of steam gauge.	No. of lbs. of coal weighed in.	No. of lbs. of ashes weighed out.	Total number of lbs. consumed.	Readings of steam gauge.	Time of filling tank.	Weight of tank full.	Weight of tank empty.	Amount of water used.	
1	10.10	438	....	....	65.0	465	....	....	67.0	483	....	....	70.0	10.00	....	....	....	....
2	10.15	....	....	....	72 5	....	....	....	74.0	....	....	....	75 0	10.15	1,500	396	1,154	....
3	10.30	...	....	...	73 0	....	....	...	74 0	....	....	....	75.0	10 30	1,530	638	882	....
4	10.45	....	....	....	72 5	....	...	....	72.5	....	....	....	74 0	10 45	1,650	953	697	....
5	11.00	280	....	....	72.0	274	....	...	72.0	....	....	....	74 0	10 59	1,525	370	1,155	....
6	11.15	...	....	....	71 0	...	....	....	72.0	....	....	...	72.0	11 13	1,580	368	1,212	....
7	11.30	....	....	....	72 0	....	....	....	72.0	...	...	....	72 0	11.27	1,595	371	1,224	....
8	11.45	...	....	...	72 0	....	....	....	71.0	....	....	....	72 0	11.40	1,641	376	1,265	....
9	12.00	413	19.0	623	71.0	441	24.5	672.0	70.0	498	18.5	451.5	71.0	11.55	1,630	399	1,231	....
10	12 15		.. .	... .	69.0	....	....	....	69.0	....	....	....	71 0	12.00	1,630	1,215	415	528 8
11	12.30	556	....	....	70 0	...	....	....	70 0	....	....	....	72 0	12 11	1,215	406	809	....
12	12 45		....	....	72.0	....	...	....	72 5	....	....	...	74 0	12 26	1,620	373	1,247	....
13	1.00		....	....	72 5	449	....	...	72.5	....	....	....	75.0	12.36	1,630	373	1,257	....
14	1.15		....	....	71 0	....	....	....	71.0	....	...	....	72 0	12.49	1,630	379	1,251	...
5	1.30		...	....	72 0	....	...	....	72.0	....	....	....	74.0	1.00	1,630	700	930	....
16	1.45		....	..	71 5	....	....	....	71.5	....	....	....	73.0	1.15	1,631	654	977	....
17	2.00	....	13.0	941	72 0	..	12.0	694.0	72.0	....	9.0	444.5	73.0	1.30	1 630	720	910	....
18	2.15	521	....	....	70.0	475	..	...	70.0	421	....	....	72 0	1 45	1.620	385	1,245	....
19	2.30		....	...	69 0	....	....	....	68 5	....	....	....	70 0	1 55	1,630	843	787	....
20	2 45	. .	....	....	71.0	...	....	....	71 0	....	....	....	72 5	2 00	1,577	1,173	404	472.8
21	3 00	....	...	....	72.0	..	...	....	72.0	....	....	....	74 0	2.13	1,173	380	793	....
22	3 15	478	...	....	71.0	....	....	....	71.0	....	....	....	73.0	2 26	1,630	408	1,222	....
23	3.30	....	...	....	71.0	....	....	...	71.0	....	....	....	73.0	2 30	1.631	1,281	350	....
24	3.45	....	....	...	69.0	124	....	....	69.0		....	....	71.0	2.43	1,281	396	885	....
25	4.00	...	101.5	869	73 0	590.5	92.0	827.0	73 0	...	68.5	471.0	75 0	3.00	1.630	374	1,256	....
26	4 15	265	....	....	70 0	...	....	....	70.0	412	....	....	71.0	3 14	1,632	377	1,255	....
27	4 30	...	....	....	70 0	....	....	...	70.0	....	...	....	70 0	3 28	1,638	374	1,264	....
28	4.45	398	..		71.0	....	....	....	71.0	...	...	....	72.5	3 41	1,630	388	1.242	...
29	5 00		..		72 0	367	....	..	72.0	....	...	....	74.0	3.54	1,632	377	1,255	....
30	5.15		..		72 5	....	....	....	72 5	....		....	74 0	4.00	1,528	1,143	385	457.1
31	5.30		243 0	542	73 0	....	258.0	630.5	73.0	....	115 0	315 0	74.0	4 15	1.143	375	768	....
32	.....		..		....	...	....	....	....	....	....	....	....	4 30	1,632	720	912	....
33	.....		..		....	....	....	....	....	....	....	....	....	4 40	1,620	368	1,252	....
34	. ...		..		....	..	....	....	....	....	....	....	...	4 50	1,635	377	1,258	....
35	.....		..		....	....	....	....	....	....	....	....	....	5 15	1,641	412	1 229	....
36	...	....	..		....	....	..	....	....	....	....	...	....	5.30	1,645	1,014	631	406.7

CONTINUED FROM PAGE 62.

NUMBER OF OBSERVATIONS.	Counter.	Steam Gauge.	Vacuum.	Water Gauge.	Well Gauge. Ft.	Well Gauge. In.	Height in feet calculated from water gauge.	Number of revolutions per 15-minute period.	Total number foot-pounds of work done during each 15-minute period.	Total work in foot-pounds in two-hour periods.	Number of lbs. of water evaporated in two-hour periods.	Duty—per 100 pounds of coal, at the rate of ten pounds of water evaporated for one pound of coal.
1	381.954	66.0	25.0	65	8	10.0						
2	382,171	72.0	24.5	63	8	9.6	171.8	217	121,100,000			
3	382.406	72.5	23.5	64	8	9.1	170.6	235	130,200,000			
4	382.652	71 0	26.5	66	8	8.9	174.2	246	149,133,000			
5	382.916	71 0	26 5	65	8	8.8	175.8	2·4	150,366,000			
6	383.143	70.0	26.5	69	8	7.8	178.9	227	131,866,000			
7	383,395	70 0	26.6	68	8	7.4	182.2	252	149,067,000			
8	383,648	69.5	26 6	67	8	6.0	179.2	253	147,200,000			
9	383 890	69.0	26.6	67	8	5.0	179.1	242	140,730.000	1,119,662,000	9,245	121,100,000
10	384,118	69.0	26.4	67	8	4 8	179 2	228	132,650,000			
11	384 346	70 0	26.7	66	8	4.0	178.0	228	131,770 000			
12	384.570	72 0	26.5	69	8	3.2	180.4	224	131,200,000			
13	384 791	72 0	26.5	66	8	3.0	180.2	221	129,300.000			
14	385,013	70.0	26.7	62	8	2.4	172 4	222	124,280,000			
15	385,243	70.0	26.5	64	8	1 1	170.3	230	127,170.000			
16	385.467	70.5	26 5	64	8	0.7	172.6	224	125,530,000			
17	385 693	70.5	26.9	65	8	0.0	173.8	226	127.540.000	1,029,440,000	9,817	104,800,000
18	385.919	69.0	26 7	62	7	11.0	171.5	226	125.900.000			
19	386,137	69.0	26 8	66	7	10.3	172 7	218	122,265,000			
20	386.369	70 0	27.0	69	7	10.0	180 9	232	136,269.000			
21	386,611	70.5	26.8	67	7	9.5	182.1	242	143,100,000			
22	386 824	70.0	26.7	67	7	9.2	179 8	213	124,370.000			
23	387.046	70.0	26 7	68	7	8.8	181.0	222	130.500.000			
24	387.273	69.0	24.9	63	7	8.2	176.4	227	130,035,000			
25	387,519	73.0	25.0	69	7	7.1	177.7	246	141,933,000	1,054,372,000	9,907	106,400,000
26	387,738	70.0	26.6	62	7	7.0	176.5	219	125,500,000			
27	387,968	69 0	26.6	114	7	4.8	228 6	230	170.700.000			
28	388,189	70.0	26.5	67	7	4.2	234.5	221	168,300,000			
29	388,429	71.5	27.4	63	7	4.0	175.6	240	136,840.000			
30	388 635	72.0	27.4	66	7	4 0	174 5	206	116.740.000			
31	388,843	72.0	27.4	67	7	4.0	179.1	208	120,970 000	839,040,000	6,050	138,700,000

SUMMARY.

Number of hours test without fire pressure	6.
" " " with " "	7½.
" pounds of coal consumed in 6 hours' test	5,993.
" " " " " 7½ " "	7,580.
" " ashes produced " 7½ " "	1,245.
" " water evaporated " 6 " "	28,969.
" " " " " 7½ " "	35,019.
Number of pounds of water evaporated for 100 pounds coal consumed in 6 hours test	483.4
Number of pounds of water evaporated for 100 pounds coal consumed in 7½ hours' test	468.1
Number of gallons of water pumped per revolution	389.8
" revolutions in 6 hours' test	5,565.
" " " 7½ " "	6,889.
Total number of gallons pumped in 6 hours' test	**2,169,237.**
" " " " " 7½ "	**2,685,332.**
Gauge number of gallons per hour for 6 hours' test	361,539.5
" " " " " 7½ " "	358,044.3
Duty from 6 hours' test	**110,600,000.**
" " 7½ " "	**115,400,000.**

In addition to the observations recorded above, a large number of indicated diagrams from the steam cylinders and from the pumps were taken by means of instruments kindly furnished by Professor Robinson, Mr. Curtis, of the P., C. & St. L. R'y, Mr. F. W. Holly, of the Holly Manufacturing Co., and Mr. Lanphear, then a student in the Ohio State University. These cards constitute a very interesting series, which will doubtless give much valuable information concerning the working of the machine, when they are properly worked up. At the date of preparation of this report this has not been done, on account of lack of time. The results, however, could have nothing to do with the question to be settled by these trials, as the conditions of the contract assumed a certain action of the pumps, as well as a definite evaporation per pound of fuel. The assumption is that the pumps throw their full capacity at every stroke, and that the fuel is consumed at the rate of one pound of coal to ten pounds of water evaporated. On this basis the duty of the engine as shown above is calculated. The rate of evaporation assumed is not excessive, and has often been exceeded. That the actual evaporation (which was determined by weighing the water on its way to the boiler) was not more than half this amount per pound of coal is no fault of the engine, but indicates a lack of efficiency in the boilers.

Concerning the slip in the pumps, it may be assumed to be less than two and one-half per cent., as it was actually found to be in a trial of a precisely similar engine at Memphis in 1882, by Mr. John W. Hill. In the above calculations no allowance is made for frictional resistance. Under similar contracts the allowance for this has been required and made by adding two pounds to the indicated pressure. If this were done, it would doubtless fully compensate for the loss for the slip in the pumps.

The duty, as shown for the last period of one hour and a half, is doubtless somewhat too high, undue weight being given to the high pressure readings during the brief period in which fire pressure was maintained. The results, taken altogether, conclusively prove, however, that the performance of the engine was about 35 per cent. above that guaranteed by the terms of the contract.

I wish to express my personal obligations to your Superintendent, Mr. Royce, and to Mr. Kilroy, your Chief Engineer, as well as to the members of your Board, for the readiness and willingness with which everything was done which might facilitate the labor of those engaged in making the tests. I have the honor to be

Yours respectfully,

T. C. MENDENHALL.

BURLINGTON, N. J.

OCTOBER, 1884.

REPORT OF HENRY P. M. BIRKINBINE, C. E.

The engine on which the following test was made was of the Gaskill Compound type, but of the smallest size built, and with unjacketed steam cylinders; and the duty obtained is considered by the builders as all that could be expected from so small a machine operating at so slow a piston speed. The shortness of the trials was due to the fact that at the time they were made the supply of water was not sufficient to allow of a longer continuous run. This the engineer states to be the case in his report.

The pumping engine furnished by The Holly Manufacturing Company, of Lockport, N. Y., for the Burlington Water Works, was submitted to a test for duty and capacity October 28th and 29th, 1884.

The manner of proceeding was not as precise and accurate in every detail as could have been pursued, yet the results may be taken as sufficiently correct to enable a practical decision as to the ability of the pumping apparatus to fulfill the contract.

At the commencement of the test the engine was in operation, steam up to working pressure and the fire in a fair condition. This, the engineer of the works, Mr. John Crook, was directed to observe carefully, and also to so conduct the firing as to leave it in as near the same condition as practicable when the tests were concluded.

While there may have been a possibility of error in the judgment of your engineer as to the condition of the fire, at the beginning and close of the test, yet from his long experience as an engineer it may be accepted as practically correct.

All the coal supplied to the fire during the test was carefully weighed, the height of the water in the boiler marked on the gauge, and the water pumped into it was passed through a meter and the quantity measured.

The test could not be continued as long as it was desired, on account of the new connection with the river not being completed. The supply of water for the pumps had to be taken from the old pump-well, and this was several times interrupted by reason of the screen becoming choked by floating matter.

The results, however, are so far above the contract guarantee that these matters of possible error may be dismissed. By the terms of the contract the Commissioners could demand a continuous test of six days. This was not thought advisable, as it would have made it necessary to allow at least 1,250,000 gallons to be wasted each day, and would have required a corresponding waste of coal. If there were any doubt of the ability of the machinery to fulfill the demands of the contract, this long test might be insisted upon, but with the large margin in favor of the engine it was not deemed necessary.

The following tables exhibit the observations made during the test.

These were made at intervals of fifteen minutes, for the pressure of the steam at the engine and the water at the pumps, the vacuum, the number of revolutions of the engine, and the quantity of water furnished to the boiler.

At intervals of an hour the stage of the water in the well was measured. The temperature of the water in the river and that of the feed water were also noted.

The firing was done by your engineer, Mr. John Crook, and was most admirably conducted, the steam and water being maintained with but trifling variations, and the coal used was judiciously applied.

TABLE 1.—GENERAL OBSERVATIONS.

DATE.	TIME.	Steam Pres. Gauge at Engine.	Water Pres. Gauge in Pounds.	Vacuum.	Register of Feed Water by Meter, Cubic Feet.	Register at Engine.	Pounds of Coal Delivered at Boiler.	Depth of Water below Gauge in Feet and Inches.	TEMPERATURE.		
									Feed.	Overflow.	River.
1884. Oct. 28 . .	10.50 A. M.	78	53	27½	307	377,377	175	15′–10″	....	....	52°
	11.05	84	57	27½	310	377,865	....	......	....	....	....
	11.20	84	57	27¼	313½	378,370	....	......	....	....	....
	11.35	84	57	27½	318	378,880	....	......	....	....	....
	11.50	84	55	27½	323	379,380	....	15′–11″	....	....	....
	12.05 P. M.	82	53	27½	328	379,879	....	......	....	....	....
	12.20	83	54	27¼	332½	380,362	175	......	....	....	....
	11.35	84	54	27¼	337½	380,854	....	......	....	....	....
	12.50	84	54	27½	342½	381,348	....	17′–3″	....	....	....
	1.05 P. M.	83	55	27½	347	381,842	....	......	....	....	....
	1.20	83	55	27½	351½	382,335	....	......	....	....	....
	1.35	82	54	27½	356	382,825	175	......	....	....	....
	1.50	82	53	27½	360½	383,322	....	17′–11″	....	....	....
Oct. 28 . .	2.05	83	55	27¼	363½	383,827	....	......	....	....	....

Mean pressure in pounds on pumps		54.714
Mean depth of water below the gauge		16.729 feet.
Total number of revolutions		6,450
Total cubic feet of water fed to the boiler, as per meter		56½
Total number pounds of coal delivered at boiler	515	
Pounds of coal not consumed	105	
Pounds of coal consumed		420

TABLE 2.—GENERAL OBSERVATIONS.

DATE.	TIME.	Steam Pres. Gauge at Engine.	Water Pres. Gauge in Pounds.	Vacuum.	Register of Feed Water by Meter, Cubic Feet.	Register at Engine.	Pounds of Coal Delivered at Boiler.	Depth of Water below Gauge in Feet and Inches.	TEMPERATURE.		
									Feed.	Overflow.	River.
1884. ct. 29 . .	10.30 A. M.	80	48	27½	380¼	390,740	175	18′–2″	....	....	52°
	10.45	85	50	27½	383½	391,278	....	......	....	....	....
	11.00	84	50	27½	388	391,820	....	......	....	....	....
	11.15	83	50	27½	393	392,371	....	......	....	....	....
	11.30	83	50	27½	397¾	392,914	....	17′–0″	....	....	....
	11.45	84	50	27½	403	392,455	....	......	....	....	....
	12.00	84	50	27½	408½	394,007	....	......	....	....	....
	12.15 P. M.	87	50	27½	413	394,536	175	......	....	....	....
	12.30	85	47	27½	417	395,080	....	19′–0″	....	....	....
	12.45	84	46	27½	421½	395,592	....	......	....	....	....
	1.00	85	50	27½	427	396,135	....	......	....	....	....
	1.15	85	49	27½	431	396,658	175	......	....	....	....
ct. 29 . .	1.26	85	49	27	433	397,040	....	20′–0″	....	....	....

ean pressure on pumps	49.1535 pounds.
ean depth of water below gauge	18.5416 feet.
otal number of revolutions....................................	6,300.
otal cubic feet of water fed to boilers, as per meter...............	52.75
otal number pounds of coal delivered at boilers.............	525
ounds of coal not consumed	153
ounds of coal consumed.. —	372

The following are the calculations for evaporation, capacity and duty, and are based pon the observations as given in Tables 1 and 2, and the following data:

TABLE 1.

2° mean temperature of feed water.
2.04 pounds, weight per cubic foot.
6.5 cubic feet of water evaporated.
20 pounds of coal consumed.

EVAPORATION.

$$\frac{62.04 \times 56.5}{420} = 8.345$$

ounds of water per pound of coal.

CAPACITY.

2.125 inches, diameter of pump-plunger.
2.125 inches, diameter of pump-plunger rod.
8 inches, stroke of pump-plunger.
2° temperature of river.
2.377 pounds, weight per cubic foot.

Gallons of water per revolution is

$$\frac{(2 \times 12.125^2 \times .7854) - (2.125^2 \times .7854) \times 18}{231} \times 2 = 35.4367.$$

Pounds per revolution is :

$$\frac{(2 \times 12.125^2 \times .7854) - (2.125^2 \times .7854) \times 18 \times 62.377}{1728} \times 2 = 295.4921$$

DUTY.

54.714, mean pressure of pump gauge.
6450 revolutions.
62 377 pounds, weight per cubic foot.

$$\frac{54.714 \times 144}{62.377} = 126.309$$

feet head above gauge.
Mean lift below gauge from well, 16.729 feet.
143.038 feet, total lift.

295.4921 × 6450 = 1,905,624.045 = total pounds of water pumped.

Duty as per contract is :

$$\frac{1,905,924.045 \times 143.038}{350.526} = 777,744$$

pounds of water raised one foot per pound of coal, or **77,770,400 pounds** per 10 pounds of coal; evaporation, 10 pounds.

Duty as per actual evaporation is :

$$\frac{1,905,924.045 \times 143.038}{420} = 649,094$$

pounds of water raised one foot per pound of coal, or **64,909,400 pounds** per 10 pounds of coal.

Gallons of water pumped per minute, as per table ·
195 minutes duration of test.
6,450 revolutions made.
35.4367 gallons per revolution, as above.

$$\frac{35.4367 \times 6,450}{195} = 1,172.137$$

gallons per minute, and 1172.137 × 1440 = **1,687,877 gallons** in 24 hours.

TABLE 2.

Temperature of feed water and weight per cubic foot same as for Table 1.
52.75 cubic feet evaporated.
372 pounds of coal consumed.

EVAPORATION.

$$\frac{62.04 \times 52.75}{372} = 8.79$$

pounds of water per pound of coal.

DUTY.

Pounds of water per revolution and weight per cubic foot as before.

$$\frac{49.1538 \times 144}{62.377} = 113.4736$$

feet head above gauges on pump.
Mean lift below gauge from well = 18.5416 feet.
132.0152 feet total lift of pumps..

295.4921 × 6300 = 1,851,600.23 = total pounds of water pumped.

Duty as per contract is:

$$\frac{1,861,600.23 \times 132.0152}{327.261} = 750,958$$

pounds of water raised one foot per pound of coal, or **75,095,800** pounds per 100 pounds of coal; evaporation, 10 pounds.

Gallons of water pumped per minute as per Table.
176 minutes duration of test.
6,300 revolutions made.
35.4367 gallons per revolution, as before.

$$\frac{35.4367 \times 6300}{176} = 1,290.469$$

gallons per minute, and 1,290.469 × 1440 = **1,858.275** gallons in 24 hours.

Duty as per actual evaporation, 8.79 pounds.

$$\frac{1,861,600.23 \times 132.0152}{372} = 660,643$$

pounds per pound of coal, or 66,064,300 pounds per 100 pounds of coal.

RESUME.

Results of the trial :

Duty, as per trial	**76,435,100 pounds.**
Duty, as per contract	**65,000,000 pounds.**
Capacity, as per trial	**1,773,076 gallons.**
Capacity, as per contract	**1,500,000 gallons.**

The calculations for quantity of water delivered are based upon the capacity of the pumps, estimated by the displacement of the plunger, without any allowance for leakage or loss of any kind. As the pumps are new and the valves and packing in good condition, this will probably amount to but a small fraction. A generous allowance for loss can be made and the pumps still perform the guaranteed duty.

It will be noticed that the quantity delivered was in excess of the guarantee, but these tests do not show how much the pumps can pump, as the supply to the well through the old pump was inadequate. There will be no difficulty in running the pumps at a speed of 30 per cent. above what was done during the test and pumping a corresponding increased quantity of water. Nor is there any difficulty in exceeding the maximum pressure for fire purposes to the same extent.

Respectfully submitted,

[Signed] HENRY P. M. BIRKINBINE,
Consulting Engineer.

PHILADELPHIA, PA., October 31, 1884.

BUFFALO, N. Y.

DECEMBER, 1885.

REPORT OF JOHN W. HILL, M. E.

To the Honorable Board of Water Commissioners of the City of Buffalo: LOUIS P. REICHERT, JAMES RYAN, G. W. MILLER.

GENTLEMEN:—I have the honor to report to you upon the performance of the Gaskill Compound Pumping Engine recently erected in your lower pump-house by the Holly Manufacturing Company, of Lockport, N. Y., as follows:

The engine is of the horizontal beam rotative type, compound condensing, and is really two independent and complete engines coupled—with cranks at quarters upon the shaft—with one fly-wheel common to both.

The high-pressure steam cylinders are placed above and parallel with the low-pressure steam cylinders, with their pistons moving in opposite directions.

Each half of the engine contains one high-pressure steam cylinder, one low-pressure steam cylinder and one double-acting plunger-pump, with central packed gland.

The steam cylinders are jacketed sides and heads.

The pumps are in line with the low-pressure cylinders, with stiff connections between the low-pressure pistons and the plungers. The housings which support the engine beams are placed between the pump and steam cylinders, with the bearings for beam centers midway between the center line of high and low-pressure steam cylinders.

The upper pins of the engine beams are connected by means of short links to the pins of the cross-heads of high-pressure pistons, and the lower pins of the beams are similarly connected to the pins of the cross heads of low-pressure pistons.

The power developed in the high-pressure cylinders is transmitted through the engine beams to the pump-plungers, and the power developed in the low-pressure cylinders is transmitted to the pump-plungers direct through stiff connections.

The low-pressure steam cylinders, housings for the beam centers, and pump barrels are tied, and braced together upon the horizontal center lines of the several parts.

The dimensions of steam cylinders taken from contractor's specifications are as follows:

High-pressure cylinders	33″ diam.
Low-pressure cylinders	66″ diam.

The diameters of plungers, plunger-rods and strokes of plungers and low-pressure pistons, were carefully measured after the duty trial by the writer and found to be as follows:

Engine "A," diam. plunger, inches	34.02
Engine "A," diam. plunger-rod, inches	6.00
Engine "A," stroke of plunger and low-pressure piston, inches	47.875
Engine "B," diam. plunger, inches	34.03
Engine "B," diam. plunger-rod, inches	6.00
Engine "B," stroke of plunger and low-pressure piston, inches	47.72

The boilers supplying steam to the engine were furnished by the contractor and form a part of his general plant, and are of the following dimensions obtained from contractor's specifications:

Style of boilers, return tubular.

Number of boilers	6
Diameter of boilers, inches	66
Length of boilers, feet	18
Tubes, number	80
Tubes, diameter, inches	3½
Tubes, effective length, feet	18
Steam domes, one to each boiler, diameter, inches	33
Steam domes, one to each boiler, height, inches	42
Main steam pipe to engine, diameter, inches	10

The contractor's guarantees are:

First. A delivery of **15,000,000 U. S. Standard gallons** per diem of 24 hours, against a head of 185 feet above water pressure gauge, with a piston speed of 120 feet per minute.

Second. A duty of engine and boilers as a plant, of **100,000,000 foot pounds** for each 100 pounds of coal consumed, with a steam pressure of 80 pounds.

Third. A duty of engine and boilers as a plant, of **90,000,000 foot pounds** for each 100 pounds of coal consumed, with a steam pressure of 80 pounds, for **1800 hours** operation during a period of 90 days, as required by your printed specification dated September 27th, 1883.

The compliance of the machinery with the first and second of these guarantees it has been my duty to investigate and report on. The compliance of the machinery with the third guarantee being otherwise provided for in your printed specification noted above.

The requirements of your printed specification under which the machinery was furnished are:

First. A capacity trial of 24 hours duration.

Second. A duty trial of 48 hours to follow the capacity trial without stopping the machinery.

Third. A working trial of 1,800 hours to be accomplished within the limits of three months operation of the machinery.

The working trial to commence within two weeks of the completion of duty and capacity trials.

The capacity trial of 24 hours duration began at 10.30 A. M. Tuesday, December 1st, and terminated at 10.30 A. M. Wednesday, December 2d, with the following results:

Counter on engine, 10.30 A. M., December 1st.	378,079.
Counter on engine, 10.30 A. M., December 2d	400,078.
Revolutions on engine in 24 hours	21,999.
Revolutions of engine per hour	916.625
Revolutions of engine per minute	15.27708

Mean area plunger " A " engine :

$$34.02^2 \times 0.7854 - \frac{6^2 \times 0.7854}{2} = 894.854 \text{ sq. inches.}$$

Mean area plunger " B " engine :

$$34.03^2 \times 0.7854 - \frac{6^2 \times 0.7854}{2} = 895.388 \text{ sq. inches.}$$

Stroke of plunger, " A " engine, feet	3.9895
Stroke of plunger, " B " engine, feet	3.9767

The printed specification requires the capacities of the pumps to be estimated by the following formula, in which the efficiency of the pumps is taken at 0.97 or with an allowance of 3 per cent. " for loss of action due to the probable imperfect filling of the pumps."

$$Q = 145.122 \times A \times N \times S,$$

in which

Q=delivery of pumps in U. S. gallons per diem of 24 hours,

145.122=a constant representing the effective capacity of pumps per sq. inch of plunger for a travel of one foot per minute for a period of 24 hours and is obtained as follows:

$$\frac{1 \times 12 \times 1440 \times 2 \times 0.97}{231} = 145.122$$

A=the sum of the effective areas of all the pump-plungers in sq. inches,

$$894.854 + 895.388 = 1790.242$$

N=average revolutions of engine or double strokes of pump-plungers, per minute,

15,27708

S = average strokes of both pump-plungers, in feet,

$$\frac{3.9895 + 3.9767}{2} = 3.9831$$

and delivery of pump during capacity trial was

$Q = 145.122 \times 1790.242 \times 15.27708 \times 3.9831 =$ **15,809,078.617 gallons** for a period of 24 hours.

But the contract requires a daily delivery of 15,000,000 gallons at a piston speed of 120 feet, or $\frac{120}{3.9831 \times 2} = 15.06365$ revolutions per minute, whence the capacity of the pumps at contract speed becomes $145.122 \times 1790.242 \times 15.06365 \times 3.9831 =$ **15,588,209.97 gallons.**

Comparing the contractor's guarantee and actual performance of the pumps, we have :

Guaranteed capacity in 21 hours at 120 feet piston speed, per minute	**15,000,000**	**gallons**
Actual capacity of pumps at a piston speed of 120 feet per minute	**15,588,209.97**	**gallons**
Excess of capacity over contractor's guarantee, per diem.	**588,209.97**	**gallons**

which is nearly four per cent. of guaranteed capacity.

Subsequent to the duty trial an extra trial for capacity was made for a period of one hour, during which time the engine was operated at an average speed of 18 revolutions per minute, corresponding to a daily delivery of

145.221 × 1790.242 × 18 × 3.9831 = **18,626,256.43 gallons,**

showing an excess above the nominal capacity of pumps of **3,626,256.43 gallons** per diem of 24 hours.

During the contract capacity trial of 24 hours, the steam and water pressure gauges were read and recorded every 15 minutes, with the following averages:

Steam pressure (at engine), pounds		78.895
Error of gauge		80.895
Less water column on gauge		2.072
Corrected guage pressure		78.823
Water pressure by gauge, pounds		79.4427
Water pressure by gauge, feet		183.3537
Center of water pressure gauge to surface of water in pump-well	pounds	10.652
	feet	24.5

The coal burned during the capacity trial of 24 hours was weighed on a sealed scale in the boiler-room in uniform charges of 275 pounds to each furnace.

The furnaces under the four boilers used for the capacity trial were fired in rotation every 15 minutes, with one extra charge of 275 pounds to No. 1 furnace at 4.03 A. M. Dec. 2d, one extra charge to same furnace at 7.50 A. M. same date, and one extra charge divided equally between No. 3 and 4 furnaces at 10.08 A. M. Dec. 2d. making a total consumption for 24 hours of 27,225 pounds, or at the rate of 1134.27 pounds per hour.

The trial for duty began at 10.30 A. M. Wednesday, Dec. 2d, and terminated at 10.30 A. M. Friday, Dec, 4th, embracing an uninterruped operation of the machinery for a period of 48 hours, previous to which time the machinery had operated under conditions of capacity trial for a period of 24 hours, making a total run, without break for any purpose whatever, of 72 hours.

During the duty trial the steam and water pressure gauges and the engine counter were read and recorded every 15 minutes.

The barometer, temperatures of air in engine-room, injection to condenser and hot well were read hourly.

The water levels in boilers, steam gauges on boilers and temperatures of feed water to boilers were read half-hourly.

The coal to furnace, was weighed in the boiler-room in uniform charges of 275 pounds each, and dumped in front of the boilers; and during the duty trial the furnaces and boilers, of which three were used, were fired at intervals of 15 minutes, in regular rotation.

The time of firing each charge to each furnace, the time of cleaning or slicing the respective fires, and the weights of ash and clinker from the furnaces were regularly entered in the coal record by an assistant, stationed in the boiler-house.

During the 48 hours of duty trial there were small quantities of coal thrown back at times from the regular charges weighed to the boilers, at the discretion of the contractor, to be used in case of neccessity after the cleaning of fires.

The net accumulations from the quantities of coal weighed and charged to the respective furnaces was weighed back and credited in the coal record at the end of trial.

The coal burned was of the quality furnished the pump-house under the contract of your Board with the coal dealers.

The particular coal burned during the trials was from the mines of the Philadelphia and Reading R. R. Coal Co.

This was fired by the regular fireman on watch during the trials, under the direction of the contractor.

The difference of surface of water in pump-well and center of water pressure gauge was read and recorded hourly from a float gauge placed in the pump-well.

The management of the machinery was in the discretion of the contractor, it being my duty to measure the work done, the fuel consumed in doing it, and to observe that the steam and water pressures and piston speed of engine complied with the contractor's stipulated conditions.

The following are the results of the trial for "duty":

Engine counter at 10.30 A. M. December 2d	400,078
Engine counter at 10.30 A. M. December 4th	433,045
Revolutions in 48 hours	42,967
Revolutions per hour	895.146
Revolutions per minute	14.9191
Piston speed per minute during 48 hours of duty trial, feet	118.848+
Mean head by water pressure gauge, 193 readings, pounds	79.700
Mean difference, center of water pressure gauge and surface of water in pump-well feet	23.7025
pounds	10.305
Contract allowance for friction of water passages in pumps, pounds	1.000
Total head as per printed specification, pounds	91.005
Coal charged to boilers No. 2, 3 and 5 during 48 hours of duty trial, 192 charges of 275 pounds	52,800
Coal weighed back at 10.30 A. M. Dec. 4th, pounds	992
Coal burned during trial, pounds	51,808
Coal burned per hour, pounds	1,079.333
Coal burned per minute, pounds	17.9889

And duty obtained as per formula of printed specification,

$$D = \frac{200 \times A \times P \times N \times S'}{C}$$

where D = duty in foot pounds per hundred pounds of coal.
A = aggregate mean area of pump-plungers = 1790.242 sq inches.
P = mean pressure per sq. inch of pump-plungers = 91.005 pounds.
N = mean revolutions, or double stroke of engine = 14.9191 per minute.
S = average stroke of both pump-plungers = 3.9831 feet,
then,

$$D = \frac{200 \times 1790.242 \times 91.005 \times 14.9191 \times 3.9831}{17.9889} = 107,638,148.482$$

foot pounds.

The average pressure in steam pipe from 193 readings of steam gauge during the duty trial was, pounds	80.276
Error of gauge, 2.000 pounds	82.276
Less water column on gauge	2.072
Corrected steam pressure, pounds	80.204

Comparing the contractor's guarantee and the actual performance of the machinery, we have:

Guaranteed duty for 100 pounds of coal burned during 48 hours of contract trial	100.000.000 foot pounds
Actual duty per 100 pounds of coal burned during 48 hours of trial.	107,638,148.482 foot pounds
Excess of duty over contractor's guarantee .... or over 7.6 per cent.	7,638,148.482 foot pounds

Upon completion of the 72 hours operation for duty and capacity trials, the bearings all around the engine were examined and none found warmed.

In addition to the data already noted for duty trial, indicator diagrams from all the steam cylinders were taken regularly every hour for 48 hours, and indicator diagrams from the pumps were taken hourly during the last 24 hours of trial.

Subsequent to the trials for duty and capacity the steam and water pressure gauges were removed from the engine and boilers and compared with an accurate test gauge for correction of errors of readings.

The general performance of the engine during the trials was very satisfactory, and I venture the opinion that the continuous operation of the machinery will be satisfactory to your Board and be beneficial to the water department.

During Dec. 7th the boilers and furnaces were tested for economy, in order to separate the work of the engine from the work of the boilers, and to compare the duty of the engine with the duties obtained from other pumping engines opon the basis of an uniform boiler economy.

The results of which, together with a more detailed report upon the trials of machinery, I will submit at an early date.

Very obediently yours,

JOHN W. HILL.

BUFFALO, N. Y., Dec. 8th, 1885.

GENERAL REPORT.

In the preliminary report, submitted December 8th, upon completion of the trials, I confined myself to the requirements of the printed specification and the guarantee of the builder in his proposition to the Water Commissioners.

In order that the record of the trials may be more fully presented to you and others interested in pumping engines of the first class, I have carefully reviewed the data taken at Buffalo, and append the following statement of dimensions of machinery and results of the several tests of engine and boilers.

DESCRIPTION OF ENGINE.

Style:—Compound condensing, beam rotative, cylinders and pumps arranged horizontally, high-pressure cylinders over low-pressure cylinders, low-pressure cylinders in line with the pumps, steam cylinders, jacketed heads and sides; pumps, double-acting plunger, central packed gland; beam arranged vertically, with end pins connected by short links with cross-heads of high and low-pressure pistons; single shaft and fly-wheel, with two cranks at quarters; one high-pressure steam cylinder, one low-pressure steam cylinder, and one pump to each half of engine.

DIMENSIONS OF ENGINE.

STEAM CYLINDERS.

H. P. cylinders (2), diameter, inches	33.
L. P. cylinders (2), diameter, inches	66.
H. P. piston rods [single], diameter, inches	4.5
L. P. piston rods [double], diameter, inches	5.
Stroke H. P. and L. P. pistons—engine "A"	47.875
Stroke H. P. and L. P. pistons—engine "B"	47.72
Steam cylinders [4] jacketed.	
Steam cylinders, heads [8] jacketed.	
Clearance H. P. and L. P. cylinders	0.025

STEAM VALVES AND PORTS.

Cut-off valves, style, double-beat puppet.	
Cut-off valves, diameter, upper seat, inches	9.125
Cut-off valves, diameter, lower seat, inches	7.875
Cut-off valves, lift, inches	0.75
Cut-off valves, area of opening, sq. inches	40.055
Intermediate valves, style, gridiron slides.	
Intermediate valves, 5 openings in seat	1″ × 13″
Intermediate valves, area of openings, sq. inches	65.
Exhaust valves, style, gridiron slides.	
Exhaust valves, 10 openings in seat	1″ × 13″
Exhaust valves, area of openings, sq. inches	130.
Steam ports H. P. cylinders	3″ × 11″
Steam ports H. P. cylinders, area, sq. inches	33.
Intermediate ports, from H. P. to L. P. cylinders	3.5″ × 20″
Intermediate ports, area, sq. inches	70.
Exhaust ports L. P. cylinders	3.5″ × 37.5
Exhaust ports L. P. cylinders, area, sq. inches	131.25

CONDENSERS AND AIR PUMPS.

Condensers, style "jet."	
Condensers, number	2.
Condensers, diameter, inches	54.
Condensers, height, inches	30.
Air pumps, style, single-acting.	
Air pumps, number	4.
Air pumps, diameter, inches	24.
Air pumps, stroke, inches	27.
Exhaust pipe to condenser, diameter, inches	14.
Injection pipe to condenser, diameter, inches	6.

STEAM PIPES.

Main steam pipe, diameter, inches	10.
Main steam pipe, length, feet	150.
Branch steam pipe [2] diameter, inches	8.

In the main steam pipe at the engine, just above the branch pipes, there is placed a quick-acting, rotative throttle valve, and in each branch pipe there is one open way screw valve.

Right angle bends in main steam pipe	4.

The steam pipe is provided with "variation" or expansion joints to compensate for alterations of length at different temperatures

BEAMS, CRANKS, SHAFT AND FLY-WHEEL.

Beams, one to each side [A and B] of engine.	
Beams, center to center of end pins, inches	76.5
Beams, center bearings	12″ × 21″
Beams, upper end pins	7″ × 17″
Beams, lower end pins	7″ × 8″
Crank pins [2]	8.5″ × 10.5″
Crank shaft, diameter, at fly-wheel, inches	16.5
Crank shaft, diameter, at bearings, inches	15.0
Fly-wheel, diameter, feet	20.0
Fly wheel, weight, tons	25.
Total weight of engine, pounds	700,000.

FEED PUMPS.

The beam shaft upon each side of engine has a prolongation—of reduced diameter—inwards, towards center line between the two sides of engine. This shaft is provided with a standard bearing near its inner end, and has keyed to it two small, equal armed beams, the larger of which works the air pumps, and the smaller works the boiler feed pumps.

Feed Pumps, style, single-acting plunger.	
Feed Pumps, number	4.
Feed Pumps, diameter, inches	6.
Feed Pumps, stroke, inches	11.

PUMPS.

Pumps, style, double-acting plunger with central packed gland.	
Pumps, number	2.
Pumps, diameter, plunger "A," inches	34 02
Pumps, diameter, plunger "B," inches	34 03
Pumps, diameter, plunger-rods [2], inches	6.
Pumps, stroke plunger "A," inches	47.875
Pumps, stroke plunger "B," inches	47.72

The pump valves are of vulcanized rubber set in composition shells, with an upwardly projecting stem guided in the hub of a cage which screws down over the outside of valve seat. The valve seats are of composition, screwed into the diaphragms of valve chambers, without bridges or ribs, and presenting an unobstructed circular waterway when the valve rises from the seat.

Pump valves, sets to each pump	4.
Pump valves, number in each set	252

Pump valves, diameter of opening in seat, inches	1.3125
Pump valves, diameter of rubber disk, inches	1.75
Pump valves, thickness of rubber disk, inches	0.5
Pump valves, lift, inches	0.3125
Pump valves, area through seats, one set, sq. inches	340.956
Pump valves, area through waterway at $\frac{5}{16}$ inch lift, sq. inches	432.952

SUCTION AND DISCHARGE PIPES.

Principal suction pipe, diameter, inches	36.
Branch suction pipe [2], diameter, inches	30.
Principal discharge pipe, diameter, inches	36.
Branch discharge pipe [2], diameter, inches	30.

Stop valves in both branches of suction and discharge pipes.

CALCULATED DATA.

Volume of high-pressure cylinder, including clearance, cu. ft	24.0237
Volume of low-pressure cylinder, including clearance and clearance of high-pressure cylinder, cu. ft	97.0265
Ratio volume of low-pressure to volume of high-pressure cylinder	4.0387
Volume of displacement, L. P. pistons [2], per hour, during 48 hours of duty trial cu. ft	336,891.4
Volume of clearance, L. P. cylinders [2], per hour, during 48 hours of duty trial cu. ft	8,422.3
Velocity of flow per second for the water through pump valves for a plunger travel of 120 feet per minute, ft	2.6628
Displacement of both pump plungers per revolution of engine, galls	742.858

BOILERS.

The boilers furnished by the contractor as part of his plant are of the return tubular type, set in independent furnaces, with independent smoke, steam and feed water connections, whereby any one or more of the boilers may be operated or laid off at will without interfering with the operation of the remainder of the battery.

The boilers are set to have three transits of the hot gas for the full length of shell, once under the shell, thence forward through the tubes, and finally back over the top of shells to the main flue at the rear end of setting. There are six boilers in the battery of the following dimensions:

Diameter of shells, inches		66.
Length of shells, feet		18.
Tubes, number		80.
Tubes, diameter, outside inches		3.5
Tubes, length, feet		18.
Grate, length of bars, inches		68.
Grate, width of furnace [each] feet		6.
Steam dome, one to each boiler, diameter, inches		33.
Steam dome, one to each boiler, height, inches		42.
Heating surface, shell	164.507	
Heating surface, tubes	1319.472	
Heating surface, heads	18.413	
Total, one boiler, sq. ft		1502.392

Super-heating surface, one boiler, sq. ft	90.206
Grate surface, one boiler, sq. ft	34.
Ratio heating to grate surface	44.188
Cross-section of tubes, sq. ft	4.261
Ratio grate surface to cross-section of tubes	7.979
Grate to boiler, front, inches	32.
Grate to boiler, back, inches	30.
Grate to boiler, floor of ash pit, inches	20.
Bridge wall concentric to boiler shell.	
Bridge wall to boiler shell, inches	6.
Chimney, cross-section, 8′ × 8″ sq. ft	64.
Chimney, height, ft	125.

In constructing the furnace fronts of the boiler setting, the contractor has provided a double set of fire doors, one immediately above the grate level, and the other about 4½ inches above the first, for charging and cleaning fires, which enables the fireman to distribute the coal over the grate to better advantage than through the usual single bank of doors at grate level, and to slice and clean fires through the lower doors without exposing the heated shell of the boiler to the chilling currents of cold air entering the furnace.

The chimney is immense for the present number of boilers, and entirely out of proportion to the number used in the trials for "capacity," "duty," and for test of economy of boilers, which were four for the first trial and three for each of the others, respectively.

During the duty trial and trial for economy of boilers with three boilers in use, a chimney 4 feet, 6 inches square, would have given better results in economy than the present stack.

I regard the stack as ample for 15 to 18 boilers of same size.

CAPACITY AND DUTY TRIALS.

The complete records of data, owing to the duration of trials and frequency of observation, are too cumbersome for publication, and are only useful as showing the uniformity of conditions under which the engine and boilers were worked.

Complete tables of data are therefore omitted, and the first and last readings and averages at the several stations of observation alone are given.

CAPACITY TRIAL.

This trial lasted 24 hours, during which interval of time the following data was taken:

Steam gauge boilers No. 1, No. 2, No. 3 and No. 4, water gauges same boilers, and temperature of feed water to boilers at beginning and end of trial and regularly at half-hourly intervals between, with the following results:

Steam pressures at 10.30 A. M. Dec. 1st, 1885:

Boilers	No. 1.	No. 2.	No. 3.	No. 4.
Pressure by gauge	87.00	83.00	81.00	83.00
Correction	−2.50	+0.5	+2.00	+1.00
Corrected pressure	84.50	83.50	83.00	84.00

Steam pressures at 10.30 A. M. Dec. 2d, 1885:

Pressure by gauge	86.00	82.00	80.00	82.00
Corrected pressure	83.50	82.50	82.00	83.00

Steam pressure average for 24 hours, 49 readings:

Boilers	No. 1.	No. 2.	No. 3.	No. 4.
Pressure by gauge	84.74	80.8	79.406	81.09
Corrected pressure	82.24	81.3	81.406	82.09

WATER GAUGES.

Levels in inches above lower nuts on glass tubes at 10.30 A. M. Dec. 1st, 1885:

Boilers	No. 1.	No. 2.	No. 3.	No. 4
Levels	7.25	5.5	5.25	5.

23.00 inches.

Levels at 10.30 A. M. Dec. 2d:

Boilers	No. 1.	No. 2.	No. 3.	No. 4.
Levels	5.75	6.00	4.75	3.75

20.25 inches.

TEMPERATURE OF FEED WATER.

Dec. 1. 10.30 A. M.	Dec. 2. 10.30 A. M.	Average. for 24 hours (49 readings).
186.0	197.0	193.75

During the capacity trial the coal was weighed in uniform charges of 275 pounds by an officer of the water department, upon a new sealed scale placed in the boiler-room, and charged to the furnace in which it was finally burned.

For this trial furnaces and boilers No. 1, No. 2, No. 3 and No. 4 were used.

From previous experiments with the plant it was found that under the customary conditions of work the consumption of coal was about 1100 pounds per hour, which divided between four furnaces made the charge to a furnace for one hour consumption 275 pounds.

The firings were fixed at regular intervals of 15 minutes, in uniform rotation, as per the following extract from the coal weigher's records:

DATE, DECEMBER 1ST, 1885,—A. M.

Time.	Furnace 1.	Furnace 2.	Furnace 3.	Furnace 4.
10.30				275
.45			275	
12.00		275		
.15	275			
.30				275
.45			275	
1.00		275		
.15	275			
.30				275

The furnaces had been fired in the following order previous to commencement of trial:

No. 1, 15 minutes before; No. 2, 30 minutes before; No. 3, 45 minutes before; No. 4, one hour.

This order of consumption was observed throughout the trial, excepting at 4 o'clock A. M. Dec. 2d, an extra charge of 275 pounds was delivered to No. 1 furnace, and at 4 15 A. M., or a quarter hour later, it took its regular charge of 275 pounds; again at 7.50 A. M., same date, this furnace (No. 1) required an extra charge of 275 pounds in addition to its regular charge at 7.15 A. M.; and at 10.08 A. M. a charge of 275 pounds was equally divided between furnaces No. 3 and No. 4, making total consumption of coal:

Regular charges, 96, of 275 pounds	26,400
Extra charges, 3, of 575 pounds	825
Total	27,225

The fires were cleaned in the following order:

Furnace No. 1.	Furnace No. 2.	Furnace No. 3.	Furnace No. 4.
Dec. 1st—6.18 P. M.	6.03 P. M.	8.48 P. M.	8.33 P. M.
Dec. 2d—12.18 A. M.	1.03 A. M.	12.48 A. M.	1.33 A. M.
" " 7.50 A. M.	8.00 A. M.	8.48 A. M.	7.33 A. M.

Ash, clinker and unburnt coal filtered through the grates were weighed back as follows:

At 1.15 A. M. December 2d	1264 pounds
At 7.43 A. M. December 2d	522 pounds
At 8.05 A. M. December 2d	905 pounds
Total	2691 pounds

Percentage of non-combustible for capacity test,

$$\frac{2691 \times 100}{27225} = 9.8843$$

The records of the engine-room consisted of the readings of engine counter, steam gauge in pipe, water gauge connected with the force main every fifteen minutes; temperature of air, injection to condensers, and overflow from hot well and barometer hourly; and the reading of vacuum gauge half-hourly, with the following initial and final readings and averages:

Readings at 10.30 A. M. Dec. 1st:

Steam pres.	Counter.	Water pres.	Vacuum.
80 Corrected, 79.928	378,079	80	25.75

Readings at 10.30 A. M. Dec. 2d:

Steam pres.	Counter.	Water pres.	Vacuum.
80 Corrected, 79.928	400,078	80	26.25

Mean for 24 hours:

Steam pres.	Counter.	Water pres.	Vacuum.
97 readings, 78.895 Corrected, 78.823	97 readings, 21,999 Revolutions.	97 readings, 79.4427	49 readings, 26.091

Revolutions in 24 hours	21,999.
Revolutions per hour	916.625
Revolutions per minute	15.27708

The initial and final readings and means for 24 hours of the hourly readings are given in the following table:

At 10.30 A. M. Dec. 1st:

Air.	Barometer.	Injection.	Hot Well.
66	29.73	44	105

At 10.30 A. M. Dec. 2d :

Air.	Barometer.	Injection.	Hot Well.
70	29.56	44	102

Mean for 24 hours:

Air.	Barometer.	Injection.	Hot Well,
25 readings,	25 readings	25 readings.	25 readings,
68.083	29.671	43.83	99.875

Recapitulation of averages and totals for boiler and engine-room during 24 hours of capacity trial:

Steam pressure at boilers—No. 1, pounds	82.24
No. 2, pounds	81.30
No. 3, pounds	81.406
No. 4, pounds	82.09
Average of all gauges, pounds	81.759
Temperature of feed water	193.75
Total coal burned, pounds	27,225.
Total ash and clinker weighed back, pounds	2,691.
Steam pressure at engine, pounds	78.823
Vacuum, inches	26.091
Water pressure by gauge, pounds	79.4427
Temperature of air, Fahr	68.083
Temperature of injection water, Fahr	43.83
Temperature of overflow from hot well, Fahr	99.875
Barometer, inches	29.671

The mean difference of center or water pressure gauge and surface of water in pump-well from random readings of the float gauge, during the capacity trial, was 24.5 feet = 10,652 pounds; and total head under which pumps worked, according to requirement of printed specification:

By water pressure gauge, pounds	79.4427
By difference water pressure gauge and surface of water in pump well, pounds	10.6520
By allowance for friction, pounds	1.0000
Total head, pounds	91.0947

and duty developed during capacity trial:

$$\frac{100 \times \overset{(A)}{895.121} \times \overset{(S)}{3.9831} \times 4 \times \overset{(P)}{91\ 0947} \times \overset{(R)}{21.999}}{\underset{(C)}{27225}} = 104{,}976{,}263.01$$

foot pounds.

During the latter part of the 24-hour capacity trial the contractor decided to use but three boilers for the trial for duty, and at 10.30 A. M. Dec. 2d, No. 1 boiler was cut off by closing the steam valve, feed cock and damper.

Under the printed specification I am of the opinion that the contractor could have selected any well-known anthracite coal for the purpose of his duty trials, and as there are several brands the combustible of which is considerably higher than that actually burned during the trial, he might have selected one of these and improved his duty accordingly.

The use of the ordinary stock coal commonly burned at the pump-house shows great faith upon the part of the contractor in the ability of his machinery to comply with the guarantees of the contract, the conditions of which, compared with most trials for high duty, would be regarded as very unfavorable.

There are several brands of antharacite, the efficiency of which when properly burned under steam boilers is 94 to 95 per cent. of the weight of the coal, and had the contractor required coal of this kind his duty would have been increased in the ratio of 95 or 94 to 90.1157, or the duty of the capacity trial would have been

$$\frac{104{,}976{,}263.01 \times 94}{90.1157} = 109{,}501{,}104.92$$

foot pounds per 100 pounds of coal.

During the extra capacity trial, which was made as a test of the possible speed of engine in case of emergency, the steam and water pressure gauges and engine counter were read every five minutes, with the following averages:

Steam pressure at engine, corrected, pounds	83.086
Water pressure, pounds	65.5
Revolutions for one hour	1080.
Revolutions per minute	18.

DUTY TRIAL.

The final readings at the different stations for the capacity trial under the requirements of the printed specification, became the initial readings for the 48-hour duty trial, and were as follows, for beginning and ending of trial and as an average for whole trial of 48 hours duration.

STATIONS.	Dec. 2, 10.30 A. M.	Dec. 4, 10.30 A. M.	Average for 48 hours.
Steam pressure at boilers—No. 2	82.5	83.5	82.817
No. 3	82.0	84.0	83.651
No. 4	83.0	84.0	83.286
Water level in boilers—No. 2	6.	6.25	
No. 3	4.75	5.5	
No. 4	3.75	4.75	
Temp. of feed water to boilers	197.0	178.	186.661
Engine counter	400.078	443.045	42.967
Steam pressure at engine	79.928	81.928	80.204
Vacuum	26.25	26.00	25.975
Water pressure	80.00	79.00	79.70
Difference levels of water pres. gauge and surface of water in pump-well	24.475	24.425 ft.	23.702
Temp. air engine-room	70.0	73.0	70.177
Temp. injection water	44.0	42.0	43.437
Temp. overflow from hot well	99.0	92.0	92.760
Barometer	29.56	29.19	29.149

The coal was burned during the 48 hours of contract duty trial and during the extra hour of trial (added to avoid a possible reduction of the fires at end of 48 hours) in the same manner as for the previous 24 hours of capacity trial, in uniform charges of 275 pounds, and firing the furnaces respectively at intervals of 15 minutes.

For the 48–49 hours of duty trial furnaces No. 2, No. 3 and No. 4 were used, burning the same coal per hour on three grates as were previously burned on four grates during the capacity trial.

The four furnaces in use for the capacity trial represented an aggregate area of grate surface of 136 square feet, and the coal burned per hour 1100 pounds, and the rate of consumption per square foot of grate surface

$$\frac{1100}{136} = 8.09 \text{ pounds per hour.}$$

The three furnaces in use for the duty trial represented an aggregate area of grate surface of 102 square feet, and the coal burned per hour for 48 hours 1079.333 pounds, and the rate of consumption per square foot of grate surface

$$\frac{11077.333}{102} = 10.562$$

pounds per hour.

For the 49 hours of duty trial the coal was burned on same grate surface for 48 hours previous, at the rate of 1079.755 pounds per hour,

or $\frac{1070.755}{102} = 10.586$ pounds per square foot of grate.

During the 48 hours of duty trial there was weighed up and charged 275 pounds of coal for every 15 minutes interval of the time, each one of the three furnaces used receiving a charge every 45 minutes.

From the charges, at the discretion of the contractor, small quantities were thrown back into a space alloted for the purpose, to be drawn upon in the event of the steam pressure going down during the operation of cleaning fires.

Portions of this coal were so used at different times, and the remainder was weighed back and deducted from gross coal charged up to 10.30 A. M. Dec. 4th. For the last hour of trial 1100 pounds were used.

The work of the machinery for this hour was not a part of the duty trial required under the printed specification.

The duty for 48 hours of duty trial developed by the formula of the printed specification is given in the preliminary report as **107,638,148.482 foot pounds** per 100 pounds of coal burned under the boilers.

The duty for the entire 49 hours of trial is deduced from the following data:

Engine counter at 10.30 A. M. Dec. 2d	400.978.
Engine counter at 11.30 A. M. Dec. 4th	443.936.
Revolutions for 49 hours	43,858.
Head pumped against, by water pressure gauge, mean of 197 readings	79.684
Difference center of water pressure gauge and surface of water in pump-well, mean of 50 readings, 23.7199 feet	10.318
Contract allowance for friction of water passages of pumps	1.000
Total head, pounds	90.997

The coal burned during 49 hours was 196 charges of 275 pounds	53,900.
Less weighed back at 10.30 A. M., Dec. 4th	992.
Net coal consumed, pounds	52,908.
Mean area of both plunges, square inches	895.121
Mean stroke of both plungers, feet	3 9831

$$\frac{895.121 \times 3.9831 \times 4 \times 43858 \times 90.999}{529.08} = \mathbf{107,576,522.27}$$

foot pounds per hundred pounds of coal burned.

It is elsewhere shown in this report that the coal burned during the 48–49 hours of duty trial contained 10.367 per cent. of non-combustible, and that this coal was from the stock in the coal houses of the pumping station at time of trial: that it was the coal furnished the water department by the coal dealers under their yearly contract.

In addition to which, it was taken as it came, without attempt to improve its quality by culling out inferior specimens, dirt or stone.

Under the printed specifications, I am of the opinion that the contractor could have demanded any one of several well-known brands of anthracite coal, which from many tests of their quality have shown an efficiency of combustible of 94 to 95 per cent. of the original weights, and improved his duty accordingly.

Estimating the duty with coal showing not more than six per cent. of non-combustible, we have:

$$\frac{107,638,148.482 \times 94}{89.633} = \mathbf{112,882,375.43}$$

foot pounds per 100 pounds of coal.

It has been customary of late years to assume good boiler performance, with good coal and skilled fireman, at an evaporation of ten pounds of steam per pound of coal from temperature of feed water, and I have made several trials of steam-boilers with anthracite and bituminous coal, with temperatures of feed water at 180 Fahrenheit or more, and in one instance with a temperature of feed water of 107 Fahrenheit, and obtained over ten pounds net steam per pound of coal burned.

In several modern instances of contracts for pumping engines, where the construction of boilers was not a part of the contract for engine, or where new engines for old works having sufficient boiler capacity have been contracted for, the guaranteed duty has been based upon an evaporation by the boilers of ten pounds net steam per pound of coal, or the duty instead of being so many foot pounds per hundred pounds of coal, is stated as so many foot pounds per 1,000 pounds of steam delivered to the engine.

Your previous contract with Henry R. Worthington for the 15,000,000 gallon, compound, direct-acting and duplex-pumping engine, the contract trials of which I had the honor to make in July, 1882, contained a guarantee of 70,000,000 foot pounds duty per 100 pounds of coal burned upon an evaporation by the boilers of ten pounds of steam per pound of coal.

The contract of E. P. Allis & Co. with the City of Allegheny, Pa., contained a guarantee of 95,000,000 foot pounds duty per 100 pounds of coal burned upon an evaporation by the boilers of ten pounds of steam per pound of coal.

The contract of E. P. Allis & Co. with the City of Decatur, Ill., contained a guarantee of 80,000,000 foot pounds duty per 100 pounds of coal burned upon an evaporation by the boilers of ten pounds of steam per pound of coal.

The contract of Robt. Wetherill & Co. with the City of Philadelphia, contains a guarantee of 95,000,000 foot pounds duty per 100 pounds of coal burned with ar evaporation by the boilers of ten pounds of steam per pound of coal.

The contract of Robt. Wetherill & Co. with the City of Cincinnati, contains a guarantee of 90,000,000 foot pounds duty upon actual delivery of water into Eden reservoir per 100 pounds of coal burned upon an evaporation by the boilers of ten pounds of steam per pound of coal.

In the recent trials of the engine constructed by E. P. Allis & Co. for the Hannibal, Mo., Water Company, the duty per 100 pounds of coal is based upon an evaporation by the boilers of ten pounds of steam per pound of coal.

Assuming the contract for this engine contained a guarantee of duty upon an evaporation by the boilers of ten pounds of steam to one of coal burned, then we should have had a much better duty of the engine *per se*.

The duty of the plant per hundred pounds of coal burned, in the several trials, for capacity, duty and for performance of boilers, other conditions alike, will vary as the economy of boilers, or as the weight of steam from temperature of feed water per unit of fuel.

The duty upon the actual consumption of coal during the test of boilers was **110,275,188.85 foot pounds**, with an evaporation from temperature of feed water of 8.7584 pounds of steam per pound of coal. Comparing this duty with the duty obtained during the 48 hours of duty trial, it is fair to infer the evaporation per unit of feed from temperature of feed water to have been less for the duty trial and less in proportion to the "duties" of the two trials.

Upon this assumption the steam per pound of coal from temperature of feed for the duty trial was

$$\frac{107,638,148. \times 8.7548}{110,275,188.85} = 8.549$$

pounds; and duty of engine or plant upon an evaporation by the boilers of ten pounds of steam per pound of coal would have been

$$\frac{107,638,148.5 \times 10}{8.549} = \mathbf{125,907,297}$$

foot pounds.

As an evidence of the quality of the engine considered independently of the boilers which you have purchased, I have brought the duty of it in comparison with the duties of several other celebrated pumping engines upon a uniform basis of ten pounds of steam per pound of coal.

ENGINE.	DUTY.
Gaskill, compound condensing, rotative beam engine, double-acting plunger-pumps, cylinders and heads steam-jacketed, capacity in 24 hours 15,000,000 gallons. Buffalo, N. Y., December 2–4, 1885.	
Test 48 hours duration	**125,907,297.**
Worthington, compound condensing, duplex direct acting engine, double-acting plunger-pumps, L. P. cylinders and heads steam-jacketed, capacity in 24 hours 15,000,000 gallons. Buffalo, N. Y., July 3–4, 1882.	
Test of 20 hours duration	67,812,170.

The above engines are each of 15,000,000 gallons daily capacity and stand side by side in the same pump-house, pumping in the same service.

ENGINE.	DUTY.
Reynolds, 2 engines compound condensing crank and fly-wheel, single-acting plunger-pumps, cylinders steam-jacketed, capacity (each engine) in 24 hours 6,000,000 gallons.	
Allegheny, Pa., August 30, September 4, 1884.	
Test of 24 hours duration—No. 1	106,368,339.
Test of 24 hours duration—No. 2	101,619,143.
Test of 6 hours duration—No. 2	107,009,618.
Leavitt, compound condensing, rotative beam engine, bucket and plunger-pumps, cylinders and heads steam-jacketed, capacity in 24 hours 5,000,000 gallons.	
Lynn, Mass., December 10–12, 1873.	
Test of 52 hours duration	121,153,693.
Leavitt (2), compound condensing, rotative beam engines, bucket and plunger-pumps, cylinders and heads steam-jacketed, capacity in 24 hours (each engine) 5,000,000 gallons.	
Lawrence, Mass., May 2–4, 1876.	
Test of 57 hours duration	115,748,467.
Gaskill, compound condensing, rotative beam engine, double-acting plunger-pumps, cylinders and heads steam-jacketed, capacity in 24 hours 4,000,000 gallons.	
Saratoga, N. Y., November 1–2, 1882.	
Test of 20 hours duration	107,967,037.
Hamilton, compound condensing, rotative beam engine, Simpson style, bucket and plunger-pumps, unjacketed cylinders, capacity in 24 hours 8,700,000 gallons.	
Milwaukee, May 19–21, 1875.	
Test of 48 hours duration	81,382,952.6
Quintard, 2 engines, compound condensing, rotative beam, Simpson style, bucket and plunger-pumps, cylinders unjacketed but provided with air spaces, capacity each engine in 24 hours 16,000,000 gallons.	
Chicago, April 18–22, 1876.	
Test of 48 hours duration	89,498,058
Test of 48 hours duration	85,605,774

The engines above enumerated, with the exception of the 15,000,000 gallon Worthington in your house, are all compound condensing, crank and fly-wheel engines, and represent the highest results with pumping machinery in point of economy, and with the same exception are all of modern design from the hands of eminent engineers.

From a comparison of the steam and water diagrams of "B" half of engine from 10.00 A. M., December 3, to 7.00 P. M. same date, it appears that of the total power developed in the steam cylinders, 93.2 per cent. was utilized in the work of the pumps; and a similar comparison of the steam and water diagrams of "A" half of engine, from 8.00 P. M. December 3d, to 10.00 A. M. December 4th, shows an efficiency for the pumps of 97.7 per cent. of total work, or average useful efficiency of engine 95.4 per cent.

In the following table are given the principal results from measurements of the steam diagrams:

Average pressure in steam pipes, pounds	80.204
Average vacuum, inches	29 149
Average vacuum, pounds	14 309

	"A"	"B"
Initial pressure H. P. cylinder	79.505	80.179
Cut-off in decimal of stroke, actual, H. P. cylinder	0.3531	0.3897
Terminal pressure L. P. cylinder, absolute	8.421	8.52
Counter pressure at mid-stroke, L. P. cylinder, absolute	2.238	2.372
Expansion by volumes	11.437	10.36
Expansion by pressure	11.6597	11.4254

ECONOMY OF BOILERS.

Subsequent to the trial of engine for duty the boilers were separately tested for economy during a period of 10 hours, with the general conditions of work as nearly like the duty trial as it was possible to have them.

The data of the engine-room for this trial was limited to the readings required to estimate the duty of the plant, in addition to a test of the quantity of steam expanded in maintaining the jackets of "B" engine.

The coal was weighed to the boilers in charges of 275 pounds, and the furnaces fired at regular intervals of 15 minutes, as in the duty trial.

During the latter half of boiler trial it was apparent that the coal was being supplied to the furnaces at a greater rate than was required to maintain the steam pressure, and charges were reduced first to 265 pounds each, and finally to 260 pounds each, to avoid producing better fires at ending than was had at beginning of trial.

The coal was burned at the rate of four charges an hour, beginning at 11.15 A. M., December 7th, with the first charge to No. 2 furnace, the second charge at 11.30 A. M. to No. 3 furnace, the third charge at 11.45 A. M. to No. 4 furnace, and the fourth charge at 12.00 M. to No. 1 furnace, and so on for entire trial.

There was consumed in ten hours forty charges of coal as follows:

31 charges of 275 pounds	8,525
2 charges of 265 pounds	530
7 charges of 260 pounds	1,820
Total coal	10,875
Less accumulations of coal fron the several charges during trial weighed back at 9.15 P. M.	503
Net coal burned	10,372

The fires were carefully cleaned at 11.00 A. M. and at 9.00 P. M., and ash and clinker returned at end of trial 1,070 pounds, or

$$\frac{1070}{10372} \times 100 = 10.3162$$

per cent. of original weight of coal shown as non-combustible.

The water to boilers was weighed in net charges of nominally 1,900 pounds to each, in a cask mounted on a large platform scale, and drawn into a supplemental tank connected with the feed-pumps.

In addition to the water thus weighed to boilers there was a quantity of condensation represented by the drainage from the jackets of "A" engine, which was trapped directly to the feed-pumps and could not be measured.

During the test for boiler efficiency (as already mentioned) the drainage from jackets of "B" engine was caught and weighed for entire time, and upon the assumption that the steam expended in maintaining the jackets of "A" engine was equal to the drainage caught and weighed from jackets of "B" engine, then the quantity of water weighed in the weighing cask, added to the drainage from jackets of engine "B," represented the total delivery of water to the boilers.

There was weighed and delivered to boilers 48 charges of water, as follows :

34 charges of 1,900 pounds	64,600
3 charges of 1,905 pounds	5,715
2 charges of 1,903 pounds	3,806
1 charge of 1,955 pounds.	
1 charge of 1,916 pounds.	
1 charge of 1,935 pounds.	
1 charge of 1,902 pounds.	
1 charge of 1,908 pounds.	
1 charge of 1,940 pounds.	
1 charge of 1,964 pounds.	
1 charge of 1,909 pounds.	
1 charge of 1,130 pounds.	
——	16,559
Total water from weighing cask	90,680

From 11.15 A. M. to 9.15 P. M. December 7th, there was caught and weighed from jackets of "B' engine 5,271 pounds of condensation, and total water to boilers :

From weighing tank	90,680.
From jackets "A"	5,271.
Total water, pounds	95,951.
Less waste from relief valve on feed pipe which was caught in casks and measured, pounds	1,262.8
Net water to boilers	94,688.2

And apparent evaporation per pound of coal,

$$\frac{94688.2}{10372} = 9.1292 \text{ pounds.}$$

From 39 calorimeter observations the following averages are obtained

Weight of water heated, pounds	200.
Weight of steam condensed, pounds	10.705
Initial temperature, as read	44.051
Initial temperature, as corrected	44.0513
Final temperature, as read	101.512
Final temperature, as corrected	101.6338
Elevation of heat of water by condensation of steam heat, units	57.5828
Thermal value of the steam, per pound	1177.4454
Thermal value of steam at observed pressure, 83.0	1213.33
Difference	35.8846
Latent heat of steam at observed pressure	883.8
Percentage of water entrained in the steam	4.06
Efficiency of the steam	0.9704

From the calorimeter data the net steam per pound of coal from temperature of feed water to boilers is deduced as

$$9.1292 \times 0.9594 = 8.7584 \text{ pounds.}$$

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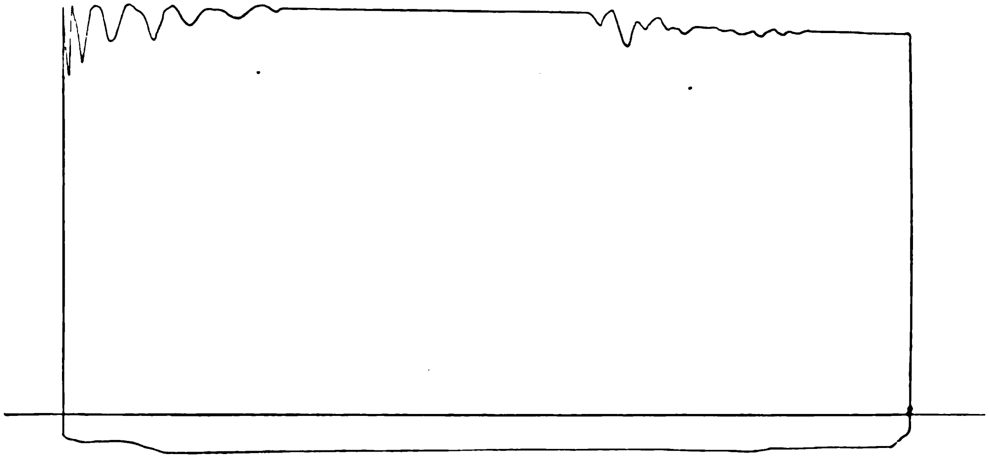
ASTOR, LENOX
TILDEN FOUNDATIONS

BUFFALO, N. Y., DEC. 3, 1885, 10 P. M.
FROM INBOARD END "A" PUMP.
Scale 40.



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These nozzles are placed out of line with the nozzle next to engine nearest the top of receiver, to prevent the steam from carrying water of condensation or primage over into the cylinders of engine.

By means of the receiver, all water of condensation or primage is intercepted before it reaches the engine, and is drawn through the jackets into the feed water heater, and finally trapped to the feed pumps, from which it is delivered back again to the boilers.

The water caught and weighed from the jackets of "B" engine during trial of Dec. 7th amounted in ten hours to 5271 pounds; assuming the consumption of steam in the jackets of "A" engine to have equaled the consumption in "B" jackets, then total consumption by jackets was 10,542 pounds.

During the same time the total steam delivered to engine was 94688.2 pounds, showing an expenditure of

$$\frac{10542}{94688.2} = 0.1113$$

of total steam in the jackets.

This large consumption of steam is accounted for in the following manner:

The intercepting receiver in the steam pipe, the steam jackets and the feed water heaters are all connected through the steam traps with the feed pumps, and through the pumps and feed pipe with the boilers, so that no loss in economy could occur by reason of an excessive flow of steam through the jackets, all heat beyond that required to properly maintain the jackets being utilized in heating the feed water in the heater, or upon mingling with the feed pumped from the hot wells into the feed pipe.

The construction of the jacket drainage and feed water apparatus is such that a loss by a large flow of steam through the jackets is balanced by a corresponding increase in the temperature of feed water to boilers, less any losses by conduction and radiation through connecting pipes, which, with the excellent methods of protection employed in this instance, must be very trifling, and perhaps considerably less than the loss in economy, which might occur with an inadequate supply of steam to the jackets.

The average temperature of drainage from jackets of "B" engine was 197.5 degrees Fahr.

During the trial for economy of boilers, sufficient data was taken in the engine-room from which to estimate the duty for the same time, as follows:

Engine counter read at intervals of 15 minutes.

Engine counter 11.15 A. M. Dec. 7th	494,740.
Engine counter 9.15 P. M. Dec. 7th	503,519.
Revolutions in 10 hours	8,779.
Head pumped against, by pressure gauge, mean of 41 readings, pounds	79.375
Difference center water pressure gauge and surface of water in pump-well, feet	25.25
pounds	10.980
Allowance for frictional resistance, pounds	1.000
Total head, pounds	91.355
The average steam pressure in pipe was, pounds	80.512
The actual coal burned was, pounds	10,372.

and duty.

$$100 \times \frac{895.121 \times 3.9831 \times 4 \times 91.355 \times 8779}{10.372} = 110,275,188.85$$

foot pounds.

The quality of coal burned during all the trials was not of the best, as shown by the large percentage of ash and clinker weighed back at end of capacity trial, at end of duty trial and for the ten hours of test of boilers.

During the first 24 hours there was burned 27,225 pounds of coal, and weighed back 2,691 pounds of non-combustible.

During the 49 hours of duty trial there was burned 52,908 pounds of coal and weighed back 5,485 pounds of non-combustible.

During the test of boilers there was burned 10,372 pounds of coal and weighed back 1,070 pounds of non combustible.

From which we derive as the percentage of non-combustible for a period of 73 hours, with a comsumption of 80,133 pounds of coal:

$$100 \times \frac{8176}{80{,}133} = 10.203 \text{ per cent. of non-combustible.}$$

For the boiler test the percentage of non-combustible is given as 10.316 per cent., and from the daily experience with this coal at the pump-house, under ordinary conditions of use, the non-combustible is stated by the engineer in charge—who weighs all the coal to the furnaces and all ash and refuse from the fires and ash-pits—to be approximately,10 per cent. of the original weights of coal.

I have appended a series of indicator diagrams from all the steam cylinders and pumps taken for different times during the duty trial to illustrate the action of the engine.

The H. P. diagrams were taken with 40 springs, the L. P. diagrams with 10 springs, and the pump diagrams with 40 springs.

In conclusion, I desire to thank the Water Commissioners and the Superintendent of Water Works, Mr. Knapp, for the excellent facilities furnished me for these trials.

Very obediently yours,

JOHN W. HILL.

CINCINNATI, Dec. 24th, 1885.

In addition to the expert trial called for in the contract with the Buffalo City Water Commissioners, a duty was required of **90,000,000 foot pounds** from the consumption of 100 pounds of coal, the engine being required to operate **1800 hours** during 90 days. The result of this long trial was an average duty of **98,232,166**, as reported by the chief engineer at the pumping station, Mr. Brennan. The average percentage of ash, which was not taken into account in calculating the above duty, was $12\frac{7}{10}$ per cent.

JACKSON, MICH.

JANUARY, 1886.

The following extract from the *Jackson* (Mich.) *Daily Citizen* of January 4th, 1886, shows the performance of a **4,000,000-gallon** Gaskill Pumping Engine recently erected at that place :

BOARD OF PUBLIC WORKS.

CHARLES CHRISTOPHER, engineer of the water works, reported in detail the thirty-six-hour test of the new Gaskill engine, built by the Holly Manufacturing Company, Lockport, N. Y. The first twelve-hour, or duty test, showed the amount of water pumped 2,113,652 gallons. Duty per 100 pounds of coal, **111,781,072 pounds** of water raised one foot high with 100 pounds of coal. The last twenty-fours run, or capacity test, was made with Jackson coal, and showed a duty of 80,816,429 pounds of water raised one foot high with 100 pounds of coal. Amount of water pumped, **4,192,904 gallons.**

The report was referred to the committee on water works, which then offered the following :

To the Board of Public Works :

GENTLEMEN—The committee on water works would respectfully submit for consideration the report of Charles Christopher, chief engineer of the water works, giving the details of the tests made on the 2d and 3d inst., of engine and boilers put up for the city at the water works by the Holly Manufacturing Company under contract dated January 26, 1885. The committee were present on the 2d and 3d while the test was being made, and witnessed the working of the engine and boilers, both while being operated by anthracite and bituminous coal. This test was only completed at 9 o'clock P. M. of yesterday, and while entirely satisfactory, yet the time is so short that we have not been able to submit tables showing the cost of pumping in as much detail as we intend to do, and such report will hereafter be submitted giving the exact cost of pumping and the saving over the work as done by the old engine and boilers, but from the experiments made and the experience from the use of the new Gaskill engine since it commenced working, and the assurance of the chief engineer, we feel justified in saying that the consumption of coal for pumping will be fifty per cent., or one-half less than with the old works. This will of itself, aside from the increased supply of water, which made the new work necessary, more than justify the city for the action taken, and the saving will go far towards paying the interest on the debt contracted. From our own observation and the report of the chief engineer we feel justified in reporting that the Holly Manufacturing Company has fulfilled its contract in the most satisfactory manner, and recommend the passage of the following resolution for the balance due on its contract.

M. SHOEMAKER,
J. L. HOLMES,
Committee.

JACKSON, MICH., Jan. 4, 1886.

Resolved, That the sum of $10,521 be audited, allowed and transmitted to the Council in favor of the Holly Manufacturing Company, for the balance due on contract dated January 26, 1885, for furnishing and erecting the engine and boilers at the water works.

The above was adopted by unanimous vote.

Superintendent Benjamin Porter reported the number of gallons pumped at the water works for the month of December at 43,277,993, a daily average of 1,396,064 gallons. There were three fire alarms but no water was used.

The Board then adjourned.

LEAVENWORTH, KAS.

JANUARY, 1886.

REPORT OF THOMAS J. WHITMAN, C. E.

ST. LOUIS, MO., Feb. 15, 1886.

L. T. SMITH, Pres't.
D. M. SWAN, Sec'y.
J. I. JONES, Supt.
Leavenworth City and Fort Leavenworth
Water Works Co.

GENTLEMEN—I herewith report the result of a trial test of the "Gaskill Compound Pumping Engine" built for you under a contract with the Holly Manufacturing Company of Lockport, N. Y.

The test was commenced at noon on the 29th of January, 1886, and it was intended to have the run of at least 24 hours duration; but owing to a want of water in the settling basins, we were compelled to stop at five o'clock A. M. of January 30th, thus reducing the trial to seventeen hours.

The shortage of water was caused by the large amount of ice on the basins. This ice also interfered greatly with the accuracy of the measurements of the quantity of water pumped, so much so, in fact, as to render useless all of the intermediate gauge readings between the commencement and end of the trial; and, indeed, to throw grave doubts upon the result of the test as regards the exact amount of water pumped.

In taking the dimensions of the ice which was floating at the commencement of the test, and was left upon the sloping walls by the receding water, I made full measurements, with the intention of being certain that when the deductions were made the quantity of water credited to the pumps would be rather less, than more, than the actual amount pumped; and I still feel very certain that the machine pumped the quantity of water I have allowed; yet there is uncertainty about the matter, and I cannot state positively that the quantity of water pumped was accurately ascertained.

The water credited to the engine as pumped has been made up by taking the quantity in the reservoirs at the beginning of the trial, and deducting the quantity in the reservoirs at the end of the trial, using the tabulated statements of reservoir quantities for the amount of water at different depths. From this quantity eleven-twelfths of the cubic contents of the ice, as figured from the measurements taken, was deducted.

The result thus obtained, compared with the calculated displacement of the pump plungers, gives a loss or slip in the pumps of $3\frac{31}{100}$ per cent. This per cent. I believe is probably in excess of the actual slip, as the pump valves are certainly well calculated to give a minimum loss in this direction.

For convenience of reference a copy of the contract is attached.

The specifications (which are declared to form a part of the contract) give the general dimensions of the various parts of the machinery to be furnished, but as one section of the contract especially states that the builders shall have the right to change any or all of these dimensions, except such as would make a change in the speed, capacity of duty, I did not consider it necessary to test the detail measurements, except as to stroke of pump and diameters of pump plungers. I note, however, that the dimensions given in the specifications for the various parts of the engines, pumps and boilers have been very closely adhered to.

One section of the contract (marked "B" in the attached copy) states: "This engine will be guaranteed to pump against two hundred pounds pressure to the square inch, and to show no signs of weakness in any of its parts while so working."

In making the test there was no attempt made to work against this extreme pressure as I considered it would simply have been straining the parts of the machine uselessly.

The last section of the specifications refers to the capacity and duty test, and reads as follows:

"Capacity.—The machine above described will be guaranteed to be capable of lifting **four million U. S. gallons** of water per twenty-four hours against a head of 369 feet, at a piston speed of 112 feet per minute, and will be guaranteed to perform a duty of **100,000,000 foot pounds** from each 100 pounds of coal, based on an evaporation of ten pounds of water per pound of coal; if a less evaporation is obtained, then the duty will be less than 100,000,000 in proportion to the reduced evaporation."

During the test the head on the pump cylinder varied from 330 feet to 344 feet, the average pressure during the run being 337 feet. This is some thirty-two feet less than the head mentioned in the specifications; but the head on the pumps at the time of the test was fully equal to the maximum head that the pumps are required to work against in regular service, and this increase of thirty-two feet could only be brought upon the pump cylinders by partially closing the valves on the delivery pipe, and as I considered that it was not desirable to throttle the delivery from the pumps in this manner, there was no attempt made to increase the pressure in the pump cylinders up to the 369 feet mentioned in the specifications.

The machine is the ordinary type of the "Gaskill Horizontal Compound Condensing" pumping machine, being two steam engines, each having a high and low-pressure steam cylinder, coupled, with cranks at 90 degrees upon the shaft, to one fly-wheel. The piston rod of the low-pressure steam cylinder is continued, and works the pump plunger located on the same horizontal line.

The pumps are two in number, each having a plunger 18 inches in diameter and 36 inches stroke. Pump rods 4 inches in diameter. These dimensions were tested and found to correspond with the specifications.

As before stated, the duration of the trial was seventeen hours, viz.: from twelve o'clock noon of the twenty-ninth of January until five o'clock A. M. of the thirtieth of January.

Revolutions made, 20,355, giving an average piston speed of 119$\frac{735}{1000}$ feet per minute.

Quantity of water pumped in seventeen hours, 3,026,045 U. S. gallons, giving a capacity of **4,272,063 U. S. gallons** in twenty-four hours, at a piston speed of 119$\frac{735}{1000}$ feet per minute: this is equal to a capacity of 4,000,000 U. S. gallons in twenty-four hours, with a piston speed of 112$\frac{11}{100}$ feet per minute; thus the machine substantially complies with the contract, as regards capacity.

Total amount of coal burned in the furnaces, 13389 pounds.

Total amount of water evaporated in boilers, 76157 pounds.

Water evaporated per pound of coal was equal to 5$\frac{688}{1000}$ pounds.

Average pressure on pump plungers, 138 pounds; equal to 318.366 feet.

Total work,—3,026,045 × 8.345 × 318.366 = 8,039,488,236 foot pounds of work.

Duty of engine:

$$\frac{8{,}039{,}488{,}236}{13389} \text{ foot pounds} \underset{\text{(coal)}}{=} 60{,}045{,}472$$

foot pounds per 100 pounds of coal.

The evaporation of the boilers was 5 $\frac{688}{1000}$ pounds of water per pound of coal. By the terms of the contract this would require a duty of 56,880,000 foot pounds per 100 pounds of coal. The duty obtained was 60,045,472 foot pounds per 100 pounds of coal, equal to a duty of **105,565,000 foot pounds** with an assumed evaporation of ten pounds of water per pound of coal.

Average measurements of the indicator diagrams taken from the pumps gives a duty of 62,840,000 foot pounds for each 100 pounds of coal burned; this would be equal to a duty of **110,478,000 foot pounds** for each 1,000 pounds of steam used.

Measuring the efficiency of the boilers by their evaporation of water from and at 212 degrees Fahrenheit, we have:

Average steam pressure of boilers, 88.14 pounds.

Average temperature of feed water, 144.8 degrees Fahrenheit.

The quantity of water actually evaporated was 76,157 pounds, equal to 6.368 pounds of water evaporated from and at 212 degrees Fahrenheit per pound of coal.

In conclusion I will state:

1st—In my opinion there is no question but that the machine has fulfilled the contract requirements as to capacity and duty.

2d—During the trial there were no indications of weakness in any part of the machinery, and generally the materials and workmanship of the engines, pumps and boilers, including all appurtenances, are good and satisfactory. In some of the details I should say a nearer approach to the requirements of the contract as stated in the section marked "D" might be had with advantage—in this, that a number of the journal bearings are not provided with brass boxes—but in all its main features I consider the machine well proportioned and strongly built, and, in my judgment, will give you satisfactory and durable service.

3d—In addition to the "tools, wrenches," etc., provided to be furnished, I would suggest that arrangements be made to have the contractor furnish a complete set of working drawings, giving detail dimensions. Such a set of drawings is almost a necessity, in order that a correct drawing of any part that may fail or require renewing, can be ready to put into the shop at once. Also to have the contractor furnish about one hundred duplicate pump valves.

4th—In regard to the section in the contract where the builders guarantee to replace, at their own expense, any part of the plant that may fail within three years (see section marked "C" in contract), your attention is called to the fact that there is no provision in the contract for the retaining of any per cent. of the contract amount as security for the performance of this part of the agreement.

I was much interested and pleased with the action of the automatic (water) governor working the cut-off motion. Under the conditions of very sudden and large variations of pressure, which, owing to local circumstances, this pumping machinery is subject to, I consider this governor a very valuable safeguard to the machinery.

Respectfully submitted,

THOS. J. WHITMAN, C. E.

KALAMAZOO, MICH.

MARCH, 1886.

REPORT OF JOHN W. HILL, M. E.

To the Honorable Common Council, Kalamazoo, Mich.: MESSRS. MCCAFFERY, WATE BURY, FULLER, *Committee on Fire and Water.*

GENTLEMEN:—I have the honor to report to you upon the contract trial of th Gaskill Compound Pumping Engine recently erected for your public water supply b the Holly Manufacturing Company of Lockport, N. Y., as follows:

The trial has been conducted in accordance with the specification and agreeme between the City of Kalamazoo and the Holly Manufacturing Company, and the usage and rules generally accepted in test trials of pumping machinery of the first class.

The agreement provides that "The Committee on Fire and Water of the City o Kalamazoo may choose an expert engineer to conduct the trial, and to ascertain if sai engine and pumps are in compliance with the contract, such choice to be approved b the Holly Manufacturing Company."

The writer has acted under this clause of the agreement. The investigation wa made to determine the compliance of the engine with the following requirements:

First. **Capacity**: The specification and agreement provides that the engine sha have a daily capacity of **3,000,000 U. S. Standard Gallons** against a domestic hea of 120 feet, and a fire pressure of 130 pounds per square inch, at a piston speed of 1 feet per minute.

Second. **Duty**: This the specification and agreement provides shall be **95,000,00 foot pounds** per 100 pounds of best coal consumed under the boiler, with a stea pressure of 70 pounds by gauge.

It is provided that the duty test "shall not be less than 24 hours" in duration.

The engine is vertical, compound, condensing, provided with a beam, crank an fly-wheels.

The steam cylinders and pumps are in line, with the former placed above the latte The pumps are of the single-acting plunger variety without side stuffing boxes plac 3′ 5″ from center to center of plungers, one plunger being driven by the H. P. stea piston and the other by the L. P. steam piston.

The working beam centers turn in bearings in the engine housings, which a located centrally between the center lines of steam cylinders, and about midwa between the steam cylinders and pumps.

The crank-shaft is bent at the center to form a pin or bearing for the outer end connecting-rod. The crank-shaft overhangs the bearings in the housings and carries fly-wheel at each end.

The crank-shaft, wheels and bearings in the engine housing are set off laterall from the steam cylinders and pumps, and the pin in the working beam for the inner e of the connecting rod set down to compensate for the angular position of the rod.

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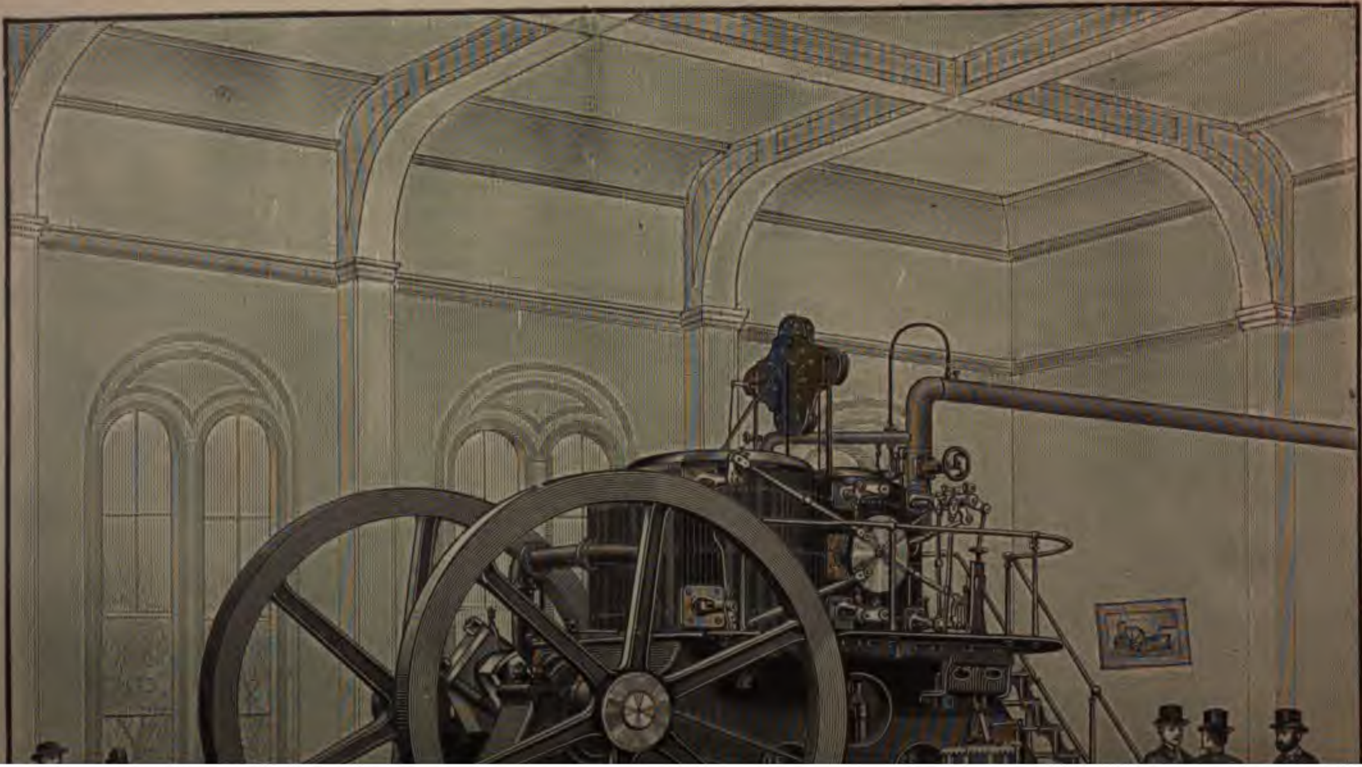

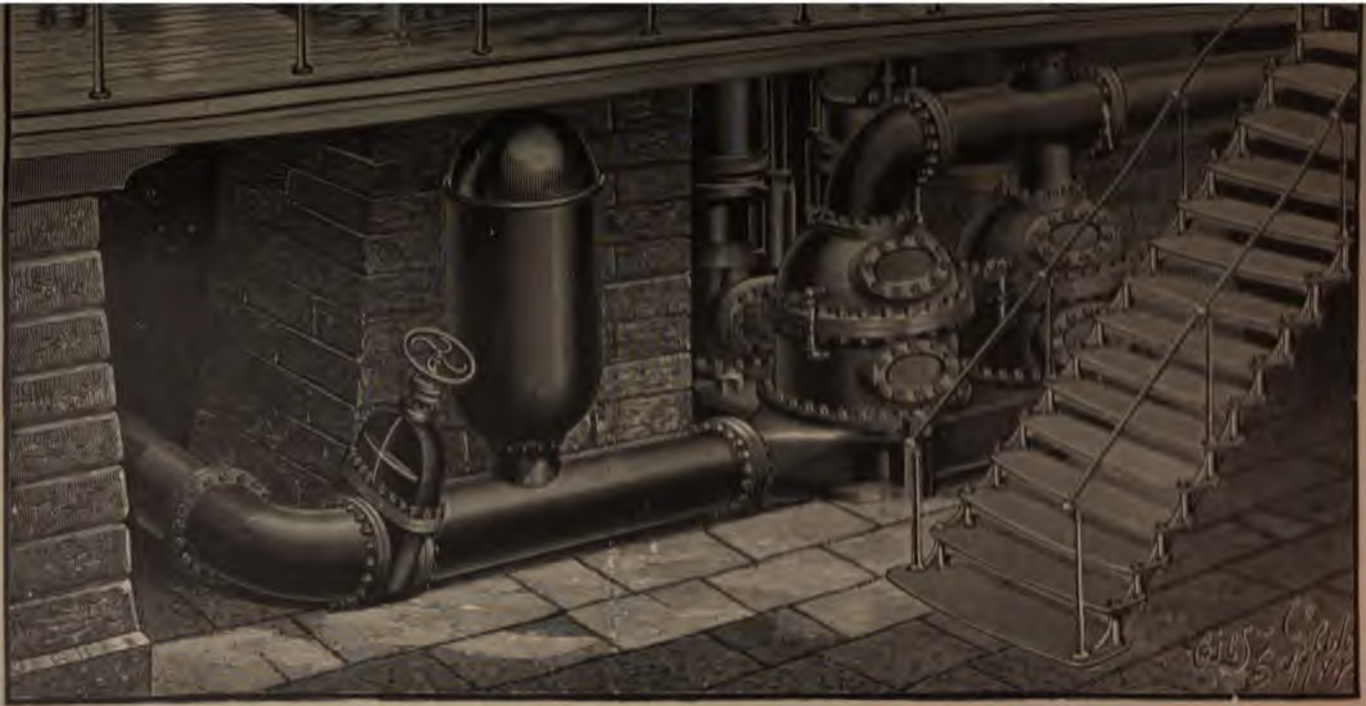

GASKILL'S VERTICAL PUMPING ENGINE, AT KALAMAZOO, MICH.

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The general design of the engine is unique in appearance, and presents many points of novelty worthy of careful study, and is another testimonial of the wonderful fecundity of the designer, Mr. Gaskill, in his line of engineering.

The design is so unlike other pumping engines that a description, in the absence of engravings, would be very unsatisfactory.

The pump plungers were carefully measured previous to the trial, with the following results:

Plunger driven by H. P. steam piston.

Diameter, inches	20.03125
Stroke, inches	29.96875

Plunger driven by L. P. steam piston.

Diameter, inches	20.03126
Stroke, inches	29.922

The other principal dimensions of the engine are as follows, taken from contractor's drawings, except steam piston rods, which were measured:

H P. steam cylinder, diameter	18.0	inches.
L. P. steam cylinder, diameter	36.0	inches.
Stroke, steam pistons (nominal)	30.0	inches.
H. P. piston rod (1), diameter	2.625	inches.
L. P. piston rod (2), diameter	3.4375	inches.
Clearance H. P. cylinder (estimated)	2.5	per cent.
Clearance L. P. cylinder (estimated)	2.5	per cent.

Steam cylinders, jacketed sides and heads, with sides lagged with walnut staves, and upper heads finished with polished covers.

Crank-shaft, diameter	12.0 inches.
Crank-pin, diameter	12.0 inches.

Fly-wheels, two in number.

Fly-wheels, diameter	10.0 feet.
Fly-wheels, weight (each)	8,000.0 pounds.

Air pumps, two single-acting, worked from pins at ends of an even lever working beam mounted on an extension of the engine beam-shaft.

Air pump, diameter	11.0 inches.
Air pump, stroke	14.0 inches

Boiler feed pumps are of the single-acting plunger style, two in number, worked from an even lever beam keyed on the extension of engine beam-shaft, outside the beam by which the air pumps are driven.

Feed pumps, diameter	3.00 inches.
Feed pumps, stroke	7.25 inches.

Pumps are fitted with four sets of small single-beat rubber valves mounted in composition shells, and working on composition seats screwed into the diaphragms of valve chambers.

Each set contains 91 valves, or 364 valves, in the suction and discharge chambers of both pumps.

Pump valves, diameter disc	1.76	inches.
Pump valves, diameter opening in seat	1.3125	inches.
Pump valves, lift	0.3125	inches.
Pump valves, thickness of disc	0.50	inches.

62609A

The valves are guided above the seat by the stem (which moves with the disc) passing through the hub in a three-winged cage, which screws down over a threaded projection of the seat outside the valve disc.

The lift of pump valve presents a clear opening between the disc and seat of 1.718 square inches, while the area through the seat (which is the full area of diameter) is 1.353 square inches.

The velocity of flow through the seats, with plungers moving at the rate of 130 feet per minute,

$$\frac{130 \times 315.1425}{123.123 \times 60} = 5.547$$

feet per second.

The pumps take water from a curbed well outside the pump-house, 20 feet diameter, 25.8 feet deep from coping. The well is supplied by infiltration through the surrounding soil, which passes into the well under the iron shoe upon which the tight curb is built.

From some data taken December 20, 1885, the well has a probable maximum inflow at the rate of four millions gallons daily. But how long this capacity can be maintained can be known only by long-continued pumping at the maximum rate.

Previous to the trials, levels were taken by Mr. Pearson, City Engineer, of the center of water pressure gauge, from which the pressures were read; of the center of the pumps, and of a bench-mark located in the pump-well, from which the stages of water in the well during the trial were taken by means of a Chesterman tape and float.

The vertical distance from the center of water pressure gauge to the center of pumps	20.48 feet.
From center of pumps to average level of water in pump-well during duty trial	14.4463 feet.
From center of water pressure gauge to bench mark in pump-well	18.000 feet.

For the duty trial, the following data was taken:

Steam pressure at boiler, every	15 minutes.
Steam pressure at engine, every	15 minutes.
Water pressure at engine, every	15 minutes.
Engine counter, every	15 minutes.
Level of water in boiler, every	15 minutes.
Vacuum in condenser, every	30 minutes.
Level of water below bench-mark in pump-well, every	30 minutes.
Temperature of feed water to boiler, every	30 minutes.
Temperature of injection to condenser	Hourly.
Temperature of overflow from air pump, every	30 minutes.
Temperature of air in engine-room	Hourly.
Indicator diagrams from steam cylinders	Hourly.
Barometer	Hourly.

The coal burned, "Lackawanna," of good quality, was weighed in uniform charges of 70 pounds on an acceptable scale in the boiler-room, at the rate of two charges an hour, and dumped just previous to firing, in front of the boiler, only one charge at a time being allowed on the floor.

The fire was worked right and left, 70 pounds to a charge every 30 minutes.

At 11.00 A. M., March 9th, the half fire to the right received a full charge of 70 pounds, and at 11.30 A. M., the half fire to the left received a charge of 70 pounds.

At 5.15 A. M., March 10th, an extra charge was divided between the right and left side of fire, and at 10.45 A. M. an extra charge was divided between the two sides of fire.

DUTY TRIAL.

The trial for duty commenced at 11.00 A. M., March 9th, and ended at 10.00 A. M., March 10th, embracing an uninterrupted operation of the engine under nearly uniform conditions of steam pressure, water pressure and speed, for a period of 24 hours. During this time the coal fired was 50 charges of 70 pounds.

Total coal fired	3,500.	pounds.
Ash, clinker and waste from ash-pit at end of trial	158.5	pounds.
Percentage of non-combustible	4.53	pounds.

The grate bars were of the rocker pattern, and these were worked frequently to clear the fire of clinker and ash during the trial. The fires were cleared of ash and clinker at the beginning of trial, and the ash-pit raked clean. The fires were of same thickness and in same condition, as nearly as could be judged from careful observation, at the beginning and at the end of trial. The condition and thickness at beginning was observed and entered in the writer's notes at the time, and the same condition and thickness was restored at the end of trial.

The average head pumped against during the trial was 59.8384 pounds, as follows:

Average pressure by water pressure gauge	42.7057	pounds.
Correction of gauge	1.0000	pounds.
Corrected gauge pressure	43.7057	pounds.

Average of difference of levels center of water pressure gauge and level of water in pump-well:

Center of water pressure gauge to bench-mark	18.00	feet.
Bench-mark to surface of water in well	16.9263	feet.
Pressure due difference of levels	15.1327	pounds.
Add allowance for frictional resistance of pumps	1.000	pounds.
Total head	59.8384	pounds.
Total head	138.106	feet.
Engine counter at 11.00 A. M., March 9th	149,986.	
Engine counter at 11.00 A. M., March 10th	188,188.	
Revolutions in 24 hours	38,202.	
Revolutions per minute	26.5292	
Piston speed per minute, feet	132.407	

Area of plungers (each)

$$20.03125^2 \times 0.7854 = 315.1425 \text{ square inches.}$$

Plunger travel per revolution of engine,

$$\frac{29.96875 + 29.922}{12} = 4.991 \text{ feet.}$$

And duty developed,

$$\frac{315.1425 \times 4.991 \times 59.8384 \times 38202}{3500} = \mathbf{102,728,884.96}$$

foot pounds.

The specifications and agreement guarantee a duty of **95,000,000 foot pounds** per 100 pounds of coal burned, which is exceeded in the actual performance of the engine by nearly **eight millions foot pounds.**

The means of the data taken during the duty trial are as follows :

Duration of trial, hours	24.	
Pressure of steam at boiler	72.062	pounds.
Pressure of steam at engine	71.287	pounds.
Level of water in boiler 11 A. M., March 9th	7.00	inches.
Level of water in boiler 11 A. M., March 10th	6.875	inches.
Level of water in boiler, mean for trial	7.0322	inches.
Vacuum by gauge observed	25.8630	inches.
Vacuum by gauge corrected	24.345	inches.
Vacuum by gauge corrected	11.951	pounds.
Temperature of feed water observed	171.344	
Temperature of feed water corrected	168.844	
Temperature of injection observed	50.937	
Temperature of injection corrected	49.937	
Temperature of overflow observed	93.156	
Temperature of overflow corrected	91.656	
Temperature of air engine-room	73.23	
Barometer, inches	29.466	
Barometer, pounds	14.465	

CAPACITY.

The contract provides that the engine shall have a capacity of **3,000,000 U. S. Standard gallons** per diem of 24 hours, at a piston speed of 130 feet per minute.

No provision is made in the contract for the measurement of the delivery of the pumps, neither are the connections of the pumps with the source of supply and the distributing mains such that an accurate measurement of the delivery can be made, except with a discontinuance of water to the city for a period of several days.

In the absence of an exact measurement of the delivery of the pumps, which measurement is not provided for by the contract, I have carefully measured the diameters and strokes of the pump-plungers, from which to deduce the probable actual delivery at contract piston speed.

The pump-plungers I carefully callipered and measured upon a Brown & Sharpe standard 24-inch steel scale, as follows :

Plunger driven by high-pressure steam piston	20.03126 inches.
Plunger driven by low-pressure steam piston	20.03125 inches.

The strokes of the plungers were measured by holding a finely-pointed lead pencil firmly down upon the upper surface of the lower cross-head, which was machine-faced, and causing it to inscribe the lineal movement of the pencil, and, by the same token, of the cross-head and connected plunger upon a dressed pine board held firmly down upon the top of pump-barrel and parallel with the axis of plunger, with the following results, measured upon the steel U. S. Standard scale :

Stroke of plunger driven by high-pressure steam piston	29.96875 inches,
Stroke of plunger driven by low-pressure steam piston	29.922 inches,
Total plunger stroke per revolution of engine	59.89075 inches,

and calculated capacity of both pumps per revolution of engine, equal to one stroke of each pump, or two single strokes of plungers,

$$\frac{20.03125^2 \times 0.7854 \times 59.89075}{231} = 81.706 \text{ gallons.}$$

The pumps being of the single-acting variety, the effective area is therefore the full area due the diameter of plungers, or

$$20.03125^2 \times 0.7854 = 315.1425 \text{ square inches};$$

and capacity of pumps per diem of 24 hours at a piston speed of 130 feet per minute,

$$\frac{315.1425}{231} \times 12 \times 130 \times 1440 = \mathbf{3,064,658.69}$$

U. S. Standard gallons.

The contractor's guaranted capacity at 130 feet piston speed per diem of 24 hours, is 3,000,000 gallons. The calculated capacity of the pumps from measured dimensions at 130 feet piston speed, per diem of 24 hours is

3,064,658.69 gallons,

allowing 64,658.69 gallons, or 2.1098 per cent. of calculated capacity, to cover the loss of action or slip of pumps.

I am of the opinion that the style of pumps furnished with your engine, with the packing around the plungers tight and the pump valves in good condition, will work with a loss of action less than found for the Memphis pumping engines; and the allowance of 64,658.69 gallons per diem in excess of the contract capacity of 3,000,000 gallons will fairly cover the probable loss of action of the pumps, and that the contract requirement for capacity has been fairly complied with.

Subsequent to the trial for duty the engine was operated for test under the provision of the specification and agreement which requires a fire pressure of 130 pounds, at 130 feet piston speed, with 70 pounds steam pressure.

To accomplish this the high-pressure cylinder was worked at full stroke, and steam direct from the pipe was taken through a 1¼-inch by-pass pipe into the low-pressure cylinder.

A few minutes after this test was commenced, a joint over the dome of the boiler in the old boiler-house blew out, preventing an operation of the engine for the full time desired.

The pressure for fire purposes is understood to mean the total head against which the pumps work, and is measured from the surface of water in pump-well, as in the duty trial.

During the test the steam and water gauges were read every two minutes by two observers, independently, and the engine counter and level of water in the pump-well similarly noted, with the following results:

Average steam pressure	68.75	pounds.
Average revolutions per minute	24.266	
Average piston speed per minute	181.111	
Average water pressure gauge, Taft	115.23	pounds.
Average water pressure gauge, Anderson	112.9	pounds.
Average, both	114.065	pounds.
Average level of water in pump-well, below bench-mark	15.427	feet.
Bench-mark below water pressure gauge	18.00	feet.
Difference surface of water in pump-well and center of water pressure gauge	33.427	feet.
Difference surface of water in pump-well and center of water pressure gauge	14.4831	pounds.
Allowance for extra frictional resistance of water passages into and out of pumps	1.000	pounds.
Total head pumped against during test for fire purposes	129.548	pounds.

With a constant steam pressure of 70 pounds, and a boiler capacity sufficient to furnish the required amount of steam, the engine—with the addition of the by-pass to take steam direct into the low-pressure cylinder—will readily work against a total pressure for fire purposes of 130 pounds from surface of water in pump-well, at 130 feet piston speed, or at the rate of 3,000,000 gallons per diem of 24 hours. In case of emergency, I believe the engine can be safely operated at a speed of 30 or more revolutions per minute.

The indicator diagrams taken during the trial have been scaled and worked, with the following average results:

Diagrams from each end of each cylinder	25.	
Average initial pressure high-pressure cylinder above atmosphere	70.49	pounds.
Average cut-off in decimal of stroke high-pressure cylinder	0.32815	
Average terminal pressure high-pressure cylinder above atmosphere.	17.54	pounds.
Average counter pressure at mid-stroke high-pressure cylinder, absolute	11.145	pounds.
Average initial pressure low-pressure cylinder above atmosphere	5.6125	pounds.
Average terminal pressure low-pressure cylinder, absolute	7.598	pounds.
Average counter pressure low-pressure cylinder, absolute	3.29	pounds.
Effective vacuum	11.175	pounds.
Mean effective pressure high-pressure cylinder, top	46.668	pounds.
Mean effective pressure high-pressure cylinder, bottom	48.452	pounds.
Mean effective pressure low-pressure cylinder, top	7.583	pounds.
Mean pressure low-pressure cylinder, bottom	9.164	pounds.
Total horse-power developed	81.891	
Ratio of expansion by pressure	11.18	
Ratio of expansion by volumes	12.03	
Coal per indicated horse-power per hour	1.757	

Comparing the moments of load of the steam piston and water plungers during duty trial, we have:

High-pressure cylinder, top,

$$254.47 \times 46.668 = 11875.6 \text{ pounds.}$$

High-pressure cylinder, bottom,

$$(254.47 - 5.412) \times 48.452 = 12067.36 \text{ pounds.}$$

Low-pressure cylinder, top,

$$1017.9 \times 7.583 = 7718.73 \text{ pounds.}$$

Low-pressure cylinder, bottom,

$$[1017.9 - (2 \times 9.28)] \times 9.164 = 9157.95 \text{ pounds.}$$

Average moment steam pistons,

$$\frac{11875.6 + 12067.36 + 7718.73 + 9157.95}{2} = 20409\ 82 \text{ pounds.}$$

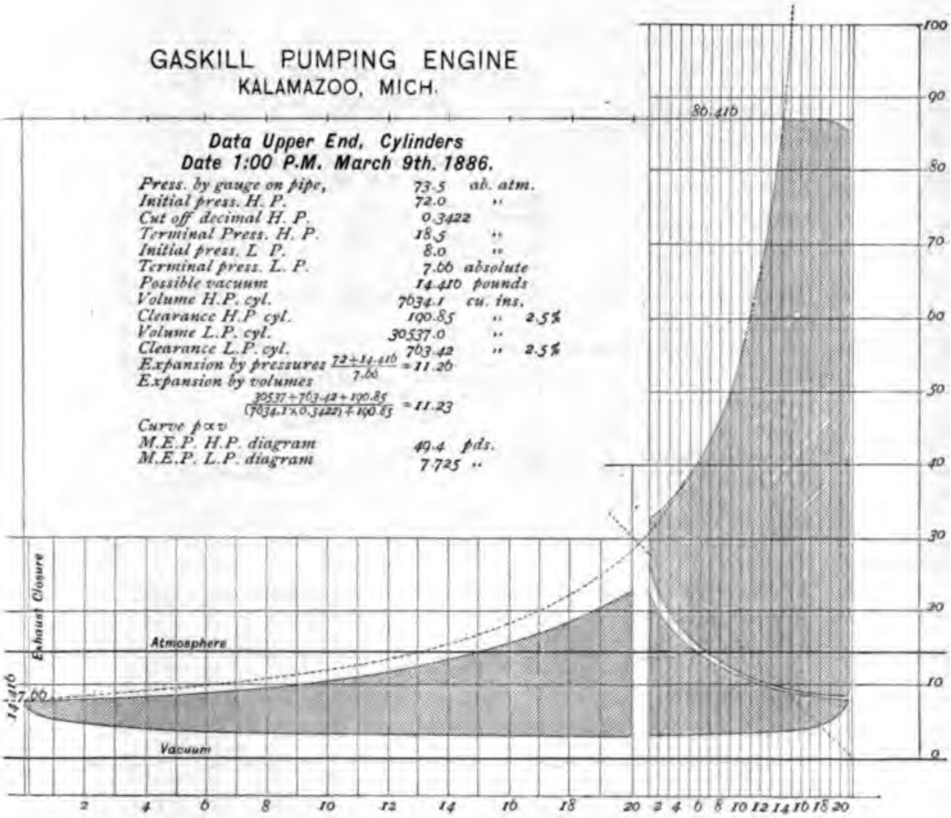
The average moment of load of water plungers during duty trial was,

$$315.1425 \times 59.8384 = 18857.623 \text{ pounds,}$$

and percentage of useful effect realized in the work of the pumps,

$$100 \times \frac{18857.623}{20409.82} = 92.395,$$

leaving 7.605 per cent. of the total power of engine to cover the losses by friction.



GASKILL PUMPING ENGINE
KALAMAZOO, MICH.

Data Upper End, Cylinders
Date 1:00 P.M. March 9th. 1886.

Press. by gauge on pipe,	73.5	ab. atm.	
Initial press. H. P.	72.0	"	
Cut off decimal H. P.	0.3422		
Terminal Press. H. P.	18.5	"	
Initial press. L. P.	8.0	"	
Terminal press. L. P.	7.66	absolute	
Possible vacuum	14.416	pounds	
Volume H.P. cyl.	7634.1	cu. ins.	
Clearance H.P cyl.	190.85	"	2.5%
Volume L.P. cyl.	30537.0	"	
Clearance L.P. cyl.	763.42	"	2.5%

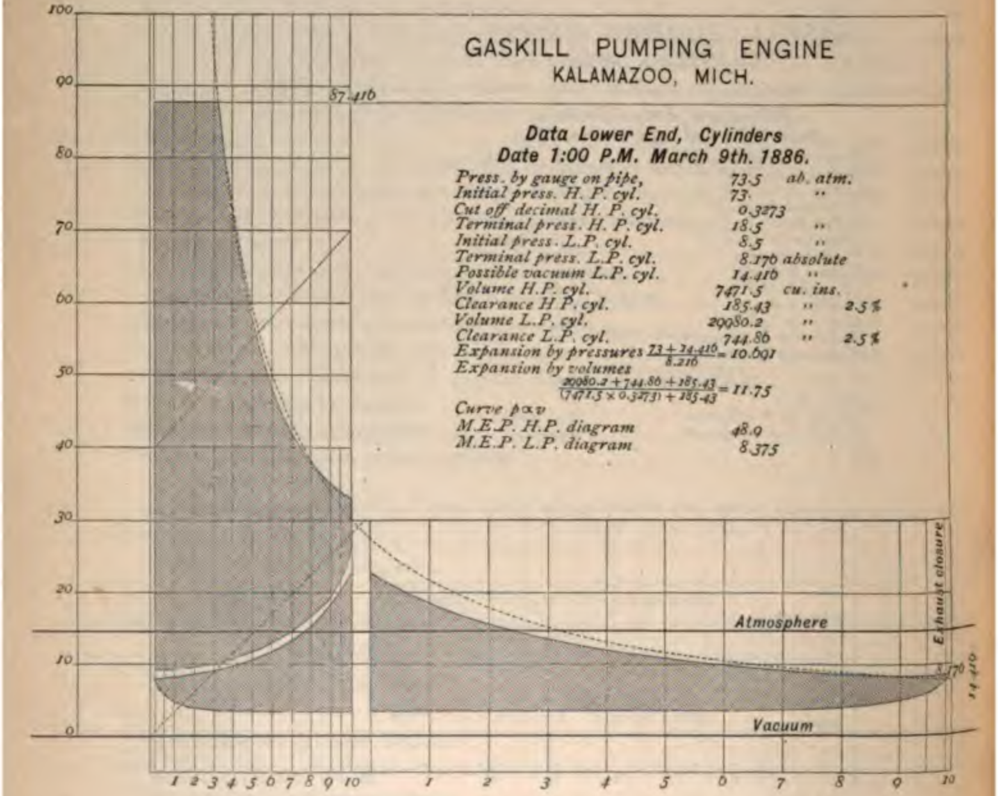
Expansion by pressures $\frac{72+14.416}{7.66}=11.26$

Expansion by volumes

$$\frac{30537+763.42+190.85}{(7634.1\times0.3422)+190.85}=11.23$$

Curve $p \propto v$

M.E.P. H.P. diagram	49.4	pds.
M.E.P. L.P. diagram	7.725	"



GASKILL PUMPING ENGINE
KALAMAZOO, MICH.

Data Lower End, Cylinders
Date 1:00 P.M. March 9th. 1886.

Press. by gauge on pipe,	73.5	ab. atm.	
Initial press. H. P. cyl.	73.	"	
Cut off decimal H. P. cyl.	0.3273		
Terminal press. H. P. cyl.	18.5	"	
Initial press. L.P. cyl.	8.5	"	
Terminal press. L.P. cyl.	8.176	absolute	
Possible vacuum L.P. cyl.	14.416	"	
Volume H.P. cyl.	7471.5	cu. ins.	
Clearance H.P. cyl.	185.43	"	2.5%
Volume L.P. cyl.	29980.2	"	
Clearance L.P. cyl.	744.86	"	2.5%

Expansion by pressures $\frac{73 + 14.416}{8.176} = 10.691$

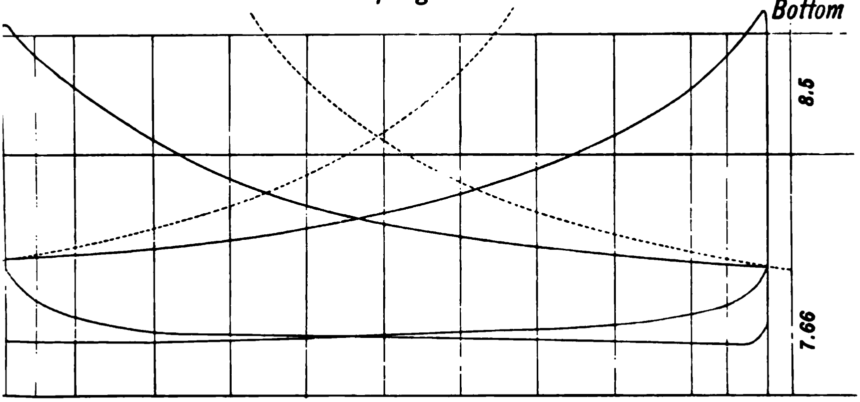
Expansion by volumes

$$\frac{29980.2 + 744.86 + 185.43}{(7471.5 \times 0.3273) + 185.43} = 11.75$$

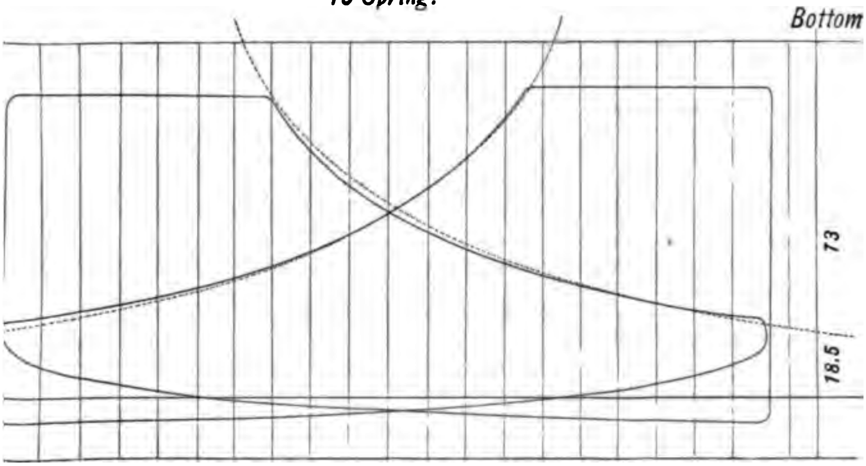
Curve $p \propto v$

M.E.P. H.P. diagram	48.9
M.E.P. L.P. diagram	8.375

Kalamazoo, Mich. 1:00 P.M. March 9, 1886.
L. P. Cylinder.
10 Spring.



Kalamazoo, Mich. 1:00 P.M. March 9, 1886.
H. P. Cylinder.
40 Spring.



The boiler furnishing steam to the engine is of the following dimensions :

Diameter shell	60.	inches.
Length of shell	16.	feet.
Number of tubes	70.	
Diameter of tubes	3.	inches.
Heating surface of shell	125.664	
Heating surface of tubes	879.648	
Heating surface of heads	16.199	
Heating surface total	1021.511	sq. feet.
Grate	4.17′ × 2.7′	
Grate surface	11.3	sq. feet.
Tube vent	2.63	feet.
Ratio heating to grate surface	90.4	
Ratio grate surface to tube vent	4.296	
Chimney (brick) 3.5′ × 3.5′	110.	ft. high.
Cross section of chimney	12.25	sq. feet.
Ratio grate surface to cross section of chimney	0.9224	

During the duty trial the coal was burned at the rate of

$$\frac{3500}{24 \times 11.3} = 11.135$$

pounds per hour, per square foot of grate surface.

No measurement was made of the water pumped to the boiler, because the duty terms of the specification and agreement required the economy of the engine to be calculated from the actual consumption of coal without reference to the economy of boiler, and the work of the boiler cannot therefore be stated with any degree of exactness ; but as an approximation, I have estimated the evaporation from the weight of steam found at release in the low-pressure cylinder, upon the assumption that eighty per cent. of the total steam would be accounted for by the indicator diagram.

Volume of low-pressure cylinder per hour,

$$\frac{1017.9 + 999.34}{144} \times 2.5 \times 26.5292 \times 60 = 55743.156 \text{ cu. feet.}$$

Volume of clearance per hour,

$$0.025 \times 55743.156 = 1393.578 \text{ cu. feet.}$$

Weight of steam per cubic foot at terminal pressure in low-pressure cylinder (7.598 absolute) = 0.0201 pound ; and weight of steam to release in low-pressure cylinder, per hour, $55743.156 + 1393.578 \times 0.0201 = 1148.448$ pounds ; less steam returned in clearance upon closure of exhaust.

Weight of steam at counter pressure in low-pressure cylinder (3.29 absolute) = 0.009328 pound.

$$1393.578 \times 0.009328 = 12.999 \text{ pounds,}$$

and steam accounted for,

$$1143.448 - 12.999 = 1135.448 \text{ pounds,}$$

and probable hourly evaporation by boiler,

$$\frac{1135.448}{0.8} = 1419.31 \text{ pounds.}$$

The coal was burned at the rate of

$$\frac{3500}{24} = 145.83 \text{ pounds per hour ;}$$

ıd steam per pound of coal,

$$\frac{1419.31}{145.83} = 9.733 \text{ pounds.}$$

The setting of the boiler and the slow rate of evaporation would suggest a very nall or no entrainment of water in the steam.

Recapitulating the results of the investigation, the engine complies fully with the ›ecification and agreement as to arrangement and proportions of the principal parts ; ill pump the guaranteed daily delivery of 3,000,000 gallons; has exceeded the ıaranteed duty by nearly 8,000,000 foot pounds, and will—with 70 pounds steam ressure—pump against a total head or pressure on the pump plungers of 180 pounds, the rate of 3,000,000 gallons per diem of 24 hours.

All of which is respectfully submitted.

Very obediently yours,

JOHN W. HILL.

CINCINNATI, March 18th, 1886.

NORFOLK, VA.

AUGUST, 1886.

REPORT OF WILLIAM WRIGHT,

CHIEF ENGINEER PUMPING STATION.

NORFOLK, VA., Aug. 12, 1886.

H. L. SMITH, ESQ., *Superintendent Water Department:*

DEAR SIR—The pumping engine built by the Holly Manufacturing Company of Lockport, N. Y., for the pumping station at Moore's Bridges is now complete with the exception of painting, and in regular operation. A duty test of the engine has recently been made, a report of which I herewith respectfully submit.

The contract between the Holly Manufacturing Company and the Board of Water Commissioners provides for one Compound Condensing Horizontal Gaskill Engine of **5,000,000 U. S. gallons** in 24 hours capacity against 40 pounds pressure, or the same capacity against a fire pressure of 133 pounds. The engine was guaranteed to pump **5,000,000 gallons** in 24 hours against 40 pounds pressure, and perform a duty of **85,000,000 foot pounds**, or, in other words, to raise 85,000,000 pounds of water 1 foot high for each 100 pounds of coal consumed under the boiler.

The engine, or rather engines, there being two distinct and separate engines coupled to the same crank or fly-wheel shaft, are of the compound-condensing crank and fly-wheel variety, so arranged that either engine may be run, or both.

The admission valves to the high-pressure cylinders are of the double-beat poppet style to which is attached a cut-off gear, adjustable by hand or controlled by a pressure regulator or governor furnished with the engine. The exhaust valves on both cylinders, high and low-pressure, are gridiron slides, worked by eccentrics fastened to a shaft in gear with the crank shaft from which the motion is transmitted to the valves. Each engine has its own air-pumps, condenser and boiler pumps. The pumps, one to each engine, are double-acting plunger pumps. The plungers work through composition rings placed central in the pumps forming the partition between the two ends.

The pumps are fitted with numerous small valves of a style used by the Holly Co. on nearly all of their pumps.

DUTY TEST.

The test trial for duty commenced at 10.33 A. M., Aug. 9th, and ended at 10.33 A. M., Aug. 10th, making an unbroken run of 24 hours' duration.

Half-hour observations were taken of steam, water and vacuum gauges, revolution counter, etc. The coal was weighed on a platform scales in a barrow furnished for the purpose, in uniform charges of 200 pounds each, and delivered to the boiler as called for by the fireman, the barrow remaining on the scales until called for.

The coal used during the test was bituminous coal, known as "Pocohontas Steam Coal," selected from the stock in the bunkers. According to the contract the test was to be made with the "best coal" The contractor was therefore at liberty to use any kind of coal he saw proper, but thinking it would give better satisfaction if the duty was obtained with the coal in use at the works, he decided to use the "Pocohontas." The coal having been hand-picked was remarkably clean, yielding less than 4 per cent. of non-combustible.

The level of the water in the boiler, and the condition of the fires were noted at the commencement of the test, and were maintained as nearly as possible in the same condition throughout the test. An extra charge of 100 pounds of coal was delivered to the boiler at end of test to bring the fires to their original condition.

In computing the duty I have made no deduction for slip, but am confident that 1 per cent. would fully cover the loss.

Diameter of H. P. cylinders		16″
Diameter of L. P. cylinders		32″
Stroke of pistons		28″
Diameter of pump plungers		19½″
Diameter of plunger rods		3¼″
Displacement in gallons per revolution		142.8
Revolutions in 24 hours		35,095.
Revolutions per minute		24.37
Piston speed in feet per minute		113.73
Average steam pressure		70 lbs.
Average water pressure		40 lbs.
Vacuum		26½″
Temperature of water in reservoir		78°
Total gallons pumped		5,011,566
Total coal consumed, pounds		4,900
Head pumped against, feet		112
40 pounds pressure=	92′	
Frictional resistance on pumps	2.3′	
Friction in suction main	1.′	
Center pressure gauge to source of supply	16.7′	
Total	112	
Per cent. of ashes		3.7
Duty in pounds raised one foot with 100 pounds of coal		95,420,216
Duty guaranteed		85,000,000
Excess above guarantee or 12¼ per cent.		10,420.216

BOILER.

The boiler in service during the test has been in service at the works for 13 years, having been furnished with the Holly Quadruplex Engine in 1873.

DIMENSIONS OF BOILER.

Style, return tubular.

Diameter of shell	60″
Length	16′
Number tubes	58

Internal diameter of tubes		3½"
Heating surface, ½ shell, square feet	125.664	
Heating surface, tubes, square feet	911.467	
Heating surface, heads, square feet	31.52	
Heating surface, total, square feet	——	1068.651
Width of grate		5′ 8″
Length of grate		6′
Grate surface, square feet		34.
Cross section of tubes, square feet		3.865
Ratio, heating to grate surface, square feet		31.43
Ratio, grate surface to cross section of tubes, square feet		8.774
Pounds of coal consumed per hour		204.17
Pounds of coal consumed per square foot of grate per hour		6.

No test of the engine under fire pressure has been made for reasons in no way reflecting upon the engine. Judging by the action of the engine July 30th, when the mains were put under 100 pounds pressure, I am confident that the engine will pump 5,000,000 gallons in 24 hours against 100 pounds pressure with both boilers in service under 70 pounds steam pressure, and that 133 pounds pressure can be gotten under like conditions with a boiler pressure of 80 pounds, but I do not deem it prudent to put old boilers such as ours under such high pressure.

I did not consider it necessary to speed the engine up to a 6,000,000 rate, as I am confident that we can safely pump that amount of water in case of an emergency

While making the test the water pumped in excess of the amount used by consumers was returned to the reservoirs by the means of a 6-inch valve and line of pipe on the force main just outside of the building.

Some difficulty was experienced during the night in keeping the engine up to speed, the demand for water being so light.

As far as the engine has been tested the specifications contained in the contract have been fully complied with in every respect.

I am confident that the engine will prove very economical under ordinary speed and pressure; but it should be borne in mind that the average duty of the engine will be considerably less than that obtained during the test, as the piston speed will be much less, and the pressure lighter.

Respectfully,

WILLIAM WRIGHT
Chief Engineer of Pumping Machinery.

CHICAGO, ILLS.

Hyde Park Pumping Station, September, 1886.

The machinery tested consisted of one Gaskill Horizontal Compound Pumping Engine, built by the Holly Manufacturing Company of Lockport, N. Y.; the following being the conditions of the contract regarding duty and capacity to be fulfilled:

"First. The engine must be of the horizontal compound condensing type, and of "12,000,000 U. S. gallons daily capacity.

"Second. The duty required on a twenty-hour test, pumping against a pressure of "60 pounds per square inch, to be not less than 100,000,000 foot-pounds, calculated "upon an evaporation of 10 pounds of water to one of coal; and at a piston speed not "exceeding 125 feet per minute; the steam pressure not to exceed 75 pounds per square "inch."

The engine is of the "compound condensing crank and fly-wheel" type, having two high-pressure and two low-pressure steam cylinders, steam jacketed on sides and heads, and two double-acting plunger pumps, connected at quarters to one common fly-wheel shaft.

Principal dimensions of engine and pumps:

Diameter of high-pressure steam cylinders	27.	inches.
Diameter of low-pressure steam cylinders	54.	inches.
Stroke of steam pistons (measured)	39.9375	inches.
Diameter of pump plungers (measured)	30.	inches.
Stroke of pump plungers (measured)	39.9375	inches.
Diameter of pump plunger rods (measured)	5.	inches.
Diameter of air pumps (4 single-acting)	20.	inches.
Stroke of air-pumps	27.	inches.
Diameter of fly-wheel	17.	feet.
Weight of fly-wheel	20.	tons.
Diameter of fly-wheel shaft	12.5	inches.

During the trial for duty and capacity, which began at two o'clock P. M., September 27th, and ended at two o'clock P. M., September 28th, readings were taken every fifteen minutes of the gauges, revolution counter, etc., in connection with the engine; and the water fed to the boilers was carefully weighed. Indicator cards were also taken from both steam and pump cylinders every hour.

From the observations recorded in the engine and boiler rooms the following results were obtained:

Average steam pressure at engine, pounds	74.76
Average vacuum in condenser, inches	26.1
Average head on water pressure gauge, feet	183.1

Vertical distance from center of water pressure gauge to floor line, feet	11.18
Average height of water in supply well below floor, feet	5.57
Friction of water in suction-main, feet	2.31
Correction of water pressure gauge, feet	1.25
Total head pumped against, feet	153.41
Revolutions of engine for 24 hours	26,167.
Gallons per revolution discharged by both pumps	481.67
Gallons of water pumped during test	**12,602,288.87**
Average piston speed, feet	121.14
Average revolutions per minute	18.17
Work performed by main pumps, foot-pounds	16,123,864,902.459
Work performed by feed pumps, foot-pounds	25,222,007.7
Effective work performed by engine, foot-pounds	16,149,087,910.159
Total water fed to boilers, pounds	146,051.
Steam used in blowing out boiler tubes, pounds	80.
Total steam used by engine, pounds	145,971.
Temperature of feed water	66° F.
Weight of one gallon of water, pounds	8.33

The capacity of the pumps per revolution is calculated as follows :

Area of plunger less one-half area of plunger rod	697.0425 inches.
Travel of both plungers for one revolution	159.75 inches.
Cubic contents of one gallon	231. inches.

$$\frac{697.0425 \times 159.75}{231} = 481.61 \text{ gallons.}$$

$$\text{Duty} = \frac{[(26167 \times 481.61 \times 8.33 \times 153.41) + 25222007.7] \times 1000}{145971} =$$

110,632,166.04 foot-pounds of effective work performed for every 1,000 pounds of steam supplied to the engine, or for every 100 pounds of coal, based on an evaporation of 10 pounds of water per pound of coal consumed under the boilers.

The water condensed from the steam supplied to the jackets of the steam cylinders was not returned to the boilers, but trapped into the sewer and wasted. No allowance was made for entrained water in the steam or for leaks in the main steam supply pipe.

The boilers were fired with "Indiana Block" coal, necessitating the use of 80 pounds of steam for the purpose of "blowing out" the boiler tubes.

The scales for weighing feed water were carefully tested before and after the trial by the official sealer of weights and measures of the Village of Hyde Park. The water pressure gauge was also carefully tested and found to indicate .54 pounds less than the actual pressure.

The level of the water in the boilers was noted at the commencement, was maintained as nearly as possible in the same condition during, and brought to the same level at the end of test.

The operation of the engine and pumps was uniform and very satisfactory throughout.

The pump valves are of the kind known as "The Troy Valve" (Gaskill's patent), being about 1.5″ diameter, and having only five-sixteenths inches lift; each pump is supplied with a sufficient number of these small valves to ensure an easy flow of water when running at full capacity.

The duty developed shows an excess over contract requirements of **10,632,166.04 foot-pounds.**

The capacity an excess of **602,288.87 gallons.**

The tests were conducted under the direction of CHAS. HERMANY, Chief Engineer of the Louisville, Ky., Water Works, assisted by J. S. Coon, M. E., on behalf of the Village of Hyde Park.

Mr. Hermany, in his preliminary report to the Board of Water Works Trustees, stated that the engine had fulfilled the contract specifications in every particular, and afterwards informally reported the duty to be about one hundred and ten million six hundred thousand foot-pounds.

BEVERLY, MASS.

MARCH, 1887.

REPORT OF WALTER H. SEARS, C. E.

BEVERLY, April 15, 1887.

MR. JOHN I. BAKER, *Chairman Water Committee:*

DEAR SIR—The pumping engines furnished and set up under contract with the Holly Manufacturing Company at the Wenham Lake Pumping Station, having been offered for the trial for capacity, arrangements were accordingly made for testing them. The water after being pumped was received from the main into a tank 16 ft. by 8 ft. by 5 ft. set outside the building, and after passing through slats and screens to prevent surging, flowed over the weir and thence back into the lake. The weir was cut from a plate of brass ¼ inch thick, with a beveled edge, and was 3 feet on the crest and 9 inches deep, and was placed in one end of the tank. The hook gauge was placed in a small box inside the tank, about four feet from the weir. Measurements were taken every five minutes and a portion of the time every minute, and the average of the readings was taken for the height on the weir. No. 1 engine was tested March 11th. The run was 5 hours, and the records for 4 hours were taken for the test. The height on the weir was 0.5 feet. The formula used was $Q = 3.326 (1 - 0.2 h) h^{\frac{3}{2}}$, in which Q represents the cubic feet per second delivered over the weir; 1 = length of crest of the weir = 3 feet; h = height on the weir = 0.5 feet; the time was 4 hours or 14400 seconds.

This gives $Q = 3.326 \times 2.9 \times 0.3535 = 3.4096489$ cubit feet per second, or 49098.944 cubic feet in 4 hours.

The displacement of the plungers per revolution is 7.1702 cubic feet; the number of revolutions in 4 hours was 6,934, making 49,718.167 cubic feet; thus showing a **loss of action of 1.24 per cent.**

No. 2 engine was tested March 17th. The run was for 5 hours, and the records of 4 hours were taken as before for the test. The height on the weir was 0.4963 feet; the number of revolutions 6820, and the **loss of action was found to be 0.96 per cent.**

The pressure against the pumps during the two tests varied from 57 pounds to 64 pounds per square inch, the suction being about 18 feet. The speed was in the vicinity of 28 revolutions per minute, or very near that required for the specified capacity of 2,000,000 gallons in 24 hours.

The contract specifies that each engine shall "be capable of lifting **two million U. S. gallons** of water per 24 hours against a head of 147 feet, and at a piston speed of 105 feet per minute." The plungers were carefully measured and found to be 13¼ inches in diameter; and this at 105 feet per minute would give **2,217,650 gallons** in 24 hours; making the deductions as above ascertained, No. 1 engine would deliver **2,190,151 gallons** in 24 hours, and No. 2 engine would deliver **2,196,359 gallons** in 24 hours.

I therefore certify that the terms of the contract in this regard are entirely fulfilled.

Very respectfully, WALTER H. SEARS,
Civil Engineer.

LIMA, OHIO.

MARCH, 1887.

REPORT OF J. D. COOK, C. E.

To the Trustees of Lima Water Works:

GENTLEMEN—Herewith I send you the result of the duty test of pumping machinery constructed by the Holly Manufacturing Company for your city.

Having no means of measuring the actual amount of water pumped, without shutting off current supply to the city, computations were made on the basis of plunger displacement, deducting 3 per cent. for slip and other defective pump action—a deduction which I deem fair and equitable to both parties.

The trial commenced at 4 o'clock P. M. on Monday, March 14th, and continued until 4 o'clock P. M. the following day, during which time careful records were taken at intervals of fifteen minutes, of counter, steam, water and vacuum gauges, and weight of feed water evaporated in the boiler—ninety readings being taken. During the entire test the performance of the machinery was uniformly satisfactory in every observed particular.

For information as to essential characteristics, details and conclusion, you are respectfully referred to the following statement, containing "description," "summary," and "result."

DESCRIPTION.

Two Gaskill Compound Duplex Crank and Fly-wheel Pumping Engines, each having:

Two H. P. steam cylinders, each 14 inches diameter.
Two L. P. steam cylinders, each 28 inches diameter.
Two pump plungers, each 15½ inches diameter.
Stroke of pistons and plungers, 24 inches.
Diameter of fly-wheel, 8 feet, 10 inches.
High pressure piston rods, 2⅛ inches diameter.
Low pressure piston rods, 2⅝ inches diameter.
Pump plunger rods, 2⅛ inches diameter.
Three return tubular boilers, each 5 feet 3 inches diameter, 16 feet long, each with 50 four-inch tubes.

One boiler fired during duty test.

SUMMARY.

Duration of test, 24 hours.
Average water pressure as shown by gauge, 64.700 lbs.
Surface of water in pump-well below gauge, 18.083 feet.

Total head or resistance used in computation, 167.542 feet.

Average steam pressure on engine gauge, 76.71 lbs.

Vacuum, 26½ inches.

Temperature of feed water, 115.

Total number of revolutions shown by counter, 41,667.

But one engine operated during test.

The actual plunger displacement per revolution, taking entire volume of plung 77.0677 gallons.

Deduction for lost pump action, 3 per cent., 2.3120 gallons.

Net pumpage per revolution, 74.7527 gallons.

Aggregate amount of water pumped, 3,114,846 gallons.

Weight of water computed at 8.344 pounds per gallon.

Feed water evaporated, 39,337 lbs.

Total coal consumed, 6,818 lbs.

Ratio of evaporation pounds of water to one pound of coal, 5.77 lbs.

Contract basis of evaporation, 10.00 lbs.

Ratio of difference between actual and contract evaporation, 1.738.

Contract duty 100,000,000 foot pounds per 100 pounds of coal on basis of evaporation of 10 pounds of water to one pound of coal.

RESULT.

Duty........... $\frac{3{,}114{,}846 \times (8.344 \times 167.542) \times 1.733}{6818} = 110{,}482{,}946$

Contract duty.......................................100,000,000

Excess..10,482,946

Respectfully submitted,

J. D. COOK,
Consulting Engineer.

TOLEDO, OHIO, March, 19th, 1887.

ERIE, PA.

APRIL, 1887.

REPORT OF FREDERICK A. SCHEFFLER, M. E.

To the Water Commissioners of the City of Erie—BENJ. WHITMAN, *President:*

GENTLEMEN—I have the pleasure of handing you my report on the duty tests of the new Gaskill Pumping Engine furnished by the Holly Manufacturing Company of Lockport, N. Y., and also in connection with the same a tabulated statement of the results obtained from the pumps, boilers and engines, which I placed in as convenient a form as I could devise for your future reference.

Throughout the tests, which were of 48 hours continuous duration, there was not the slightest difficulty with the engines. There was, of course, somewhat of a " pound " in the bearings, and pins of the connecting rods at the beginning of each stroke, but this could not very well be called a difficulty, as any large piece of new machinery, running at such a slow speed, will always need to be run somewhat loosely at the parts named until they wear closely to the proper and exact shape, when they can be adjusted and all lost motion taken up. I have no doubt but that before the engine is accepted by you the builders will have made an exceedingly quiet running engine from same.

The first test began at 7 o'clock A. M., April 26, and as each of the four tests made were of twelve hours duration, the tests concluded at 7 A. M., April 28th. Care was taken to start each test with a clean fire, note the height of water in the gauge glass at the time and to see that the fires were in the same condition at the conclusion of each test as they were when the latter began. The flues were cleaned on the mornings of the 26th and 27th, and this may account for the larger evaporation per pound of coal during the tests (Nos. 1 and 3) occurring on these two days, than that during the other two tests (Nos. 2 and 4) which occurred at night. Two boilers were found to be ample for the 5,000,000-gallon test (No. 1) and no forcing whatever was required, and the steam pressure throughout the tests was within the limit of the guarantee, *i. e.* from 70 to 80 pounds.

By reference to the tabulated statement referred to above, you will notice that have given you several duties produced by using the different divisors appearing in each of the four columns. The actual duty is shown to surpass the guaranteed duty by several millions in each test, but the net duty shows an extremely good result—exceeding **the guaranteed duty of 105,000,000 foot pounds** in the **5,000,000 gallon** test (No. 1) by **17,000,000 foot pounds,** which is nearly **17 per cent.** and in the other tests an increase of as equally good proportions. What I refer to by net duty is the duty obtained after deducting a leakage of 4½ pounds per minute of steam which leaked through the large valves communicating with the two boilers not used, the steam pipes leading to the old engines, and the blow-off pipes. Mr. Holly called my attention to these leaks sometime previous to the test, but as it would have been quite a serious undertaking to have disconnected these connections and make blank joints it was agreed that as accurate a measurement of the leaks as could be made, would be taken at the time of the tests and due allowance would be made for same.

Mr. Holly was perfectly fair and just in demanding this allowance, but it will be seen that the duty guaranteed was more than obtained without any allowance whatever. So great was the leakage into the two unused boilers that although the stop valves were closed as tightly as possible, the steam gauge on them showed 45 pounds pressure throughout the test. The weight of the leaks could be taken at only three of the drips, as the other two were so connected that it was not possible to measure them. The total leakage must have been somewhat more than 4½ pounds per minute, but Mr. Holly agreed to accept that as the average allowance. This amounted to 3,240 pounds per 12 hours. If I may be permitted to make a suggestion, I would say right here that it would be a paying investment to have new valves put in where these leaks occur, and would suggest "Jenkins" valves, the seats of which can easily be renewed, and they are always tight. This leakage of 3,240 pounds in 12 hours means a coal consumption of 1,100 pounds per 24 hours, and the new valves would soon be paid for in the saving of fuel obtained.

The coal used during the test was "Bituminous" Slack of the usual quality used by you daily. As the track scales were very convenient it was decided not to weigh the coal in small charges, but to take the gross weight at the beginning and ending of each test. This gave the weight of coal used much more accurately than could possibly have been obtained by weighing it at intervals and in small quantities. As the guaranteed duties were based on coal having an evaporative efficiency of 10 pounds water per pound, and as it was not possible to get this quality of coal, or, in fact, to burn it properly on the present grates under the boilers, the basis of the duty had to be reduced to its equivalent, which is 1,000 pounds of water.

Readings of the steam gauge, water-pressure gauge, and vacuum gauge, as well as the counter and water weighed to boilers, were taken every half hour. Indicator cards were taken twice during each test. This was hardly as often as I would liked to have had them, owing to the fact that, as there were only four indicators to be had, continual changing had to be done, which would have greatly interfered with the taking of the readings. The temperature of the feed-water was taken every two hours. It was found that this temperature was much higher during the time when the engine ran slowest, which was due probably to the fact that the water remained in the heater somewhat longer than when running more rapidly ; this suggests a larger heater which might be used to advantage. The pressure gauge on the steam jacket of the cylinders showed 60 pounds throughout the tests. No allowance was made for the water used by these jackets for the steam passed through the heater and circulated thence to the boiler, and the amount of heat imparted to the feed water was supposed to have equalized the amount of water condensed and re-evaporated in the boilers.

The pump cards do not show any slip whatsoever, and as there was no practical way at hand of finding out if there was any slip at all, and as the specifications do not say that allowance shall be made for slip during test, none was made. I believe there is a slip, but owing to the peculiar, yet very practical construction of the pump valves, this cannot be large, probably from one-half to one per cent.

All indicator cards taken show a very fine and sensitive valve mechanism, proper proportion of all parts and an extremely quick cut-off. There is very little loss of pressure of steam when passing from the high-pressure cylinder to that of the low-pressure.

To show that the allowance of 4½ pounds per minute as leakage was not too great there is an excellent proof in the indicated results, showing amount of water used by the engine.

Ninety per cent. of the water evaporated is accounted for when reckoning from the absolute terminal pressure shown on the low-pressure cards, and adding to this result the leakage referred to above.

This leaves 10 per cent. of the total water used to be accounted for, which doubtless was held in suspension in the steam. This agrees with Mr. Chas. T. Porter's report of the test of the Saratoga engines, who made this percentage $11\frac{4}{10}$ of the total amount evaporated.

I enclose you a quantity of the cards taken from the engines and pumps. If you desire a copy of the readings of each of the four tests, they will be at your disposal.

I have only to add that the engines have more than fulfilled the duty guaranteed by the contractor in all four tests.

In closing I would add that I was ably assisted during the test by Mr. S. F. Selden, of this city, who took the readings during my absence.

[Signed] Yours respectfully,

FREDERICK A. SCHEFFLER, M. E.

May 11, 1887.

TABULATED REPORT OF RESULTS

OBTAINED AT THE DUTY TEST OF THE

NEW 5,000,000 GALLON GASKILL PUMPING ENGINE

AT ERIE WATER WORKS, ERIE, PA.

THE PUMPS.

No. of test.	Date. 1887.	Duration. h.	Total Revolutions.	Average speed in feet per minute.	Gallons per 24 hours.	Average total head. feet.	Average total pressure of water. lbs.	Average water pressure per gauge. lbs.	Water evaporated in 12 hours' run. lbs.	Coal consumed in 12 hours' run. lbs.	DUTY.				Gallons waer pumped per pound of coal burned.
											Guaranteed per 1000 lbs. water. ft. pounds.	Actual duty per 1000 lbs. water. ft. pounds.	Net duty per 1000 lbs. water, allowance for leakage in pipes being 4½ pounds water per minute. ft. pounds.	Duty per 100 lbs. Bituminous slack coal. ft. pounds.	
1	Apr. 26th.	12	14536	121.128	5,298,372	244.216	105.721	92.60	47321	7575	105,000,000	113,935,453	122,309,829	71,175,450	349
2	Apr. 26-27	12	9447	78.72	3,443,431	238.256	103.141	90.02	32331	5745	90,000,000	106,042,833	117,509,567	59.503,408	299
3	Apr. 27th.	12	11840	98.64	4,315,680	238.487	103.240	90.12	38621	6355	100,000,000	111,040,929	120,926,817	67,482,482	339
4	Apr. 27-28	12	10406	86.71	3,792.987	240.566	104.054	91.02	35444	6000	95,000,000	107,481,169	117,930,208	63,313,740	316

THE BOILERS.

No. of test.	Date. 1887.	Duration. hrs.	Kind of fuel used.	Total coal burned. lbs.	Ashes and refuse. lbs.	Combustible. lbs.	Average steam pressure per gauge. lbs.	Average temperature of feed water. Deg. F.	Total water evaporated. lbs.	Equivalent evaporation from and at 212 deg. F. lbs.	Water evaporated per pound of fuel under observed conditions. lbs.	Water evaporated per pound of combustible. lbs.	Water evaporated from and at 212 deg. F. per pound of fuel. lbs.	Water evaporated from and at 212 deg. F. per pound combustible. lbs.	Pounds of water evaporated per $1.00 worth of fuel.	Gallons of water pumped per $1.00 worth of fuel.
1	April 26th.	12	Bituminous Slack.	7575	723	6852	73.84	152	47321	51878	6.24	6.92	6.84	7.56	9316	515,673
2	April 26-27.	12	" "	5745	619	5126	71.40	161	32331	35124	5.62	6.30	6.11	6.85	8300	443,823
3	April 27th.	12	" "	6355	525	5830	72.22	148	38621	42483	6.07	6.62	6.68	7.28	8965	500,903
4	April 27-28.	12	" "	6000	649	5351	72.08	170	35444	38183	5.90	6.62	6.36	7.13	8734	466,732

THE ENGINES.

No. of test.	Date. 1887.	Duration. h.	Average steam pressure. lbs.	Average vacuum. ins.	Average revolutions per minute.	Average horse-power high pressure cylinder.	Average horse-power low pressure cylinder.	Total horse-power exerted.	Horse-power accounted for in pumps.	Efficiency of engines. p. ct.	Coal burned per hour per horse-power. lbs.	Water used per hour per horse-power. lbs.	Total water evaporated by boilers. lbs.pr.h.	Water accounted for, reckoned from low pressure terminal as per indicator cards.	Water accounted for by drips (4½ lbs. per minute). lbs.	Per cent. of water accounted for.	Per cent. of steam in the state of water.
1	April 26th.	12	73.84	26	20.188	126.108	118.719	244.827	226.96	92	2.53	16.1	3943	3334	270	91	9
2	April 26-27.	12	71.40	26	13.120	77.70	76.79	154.49	143.90	93	3.09	17.4	2694	2166	270	90	10
3	April 27th.	12	72.22	26	16.44	97.70	95.71	193.41	180.50	93	2.73	16.6	3218	2714	270	92	8
4	April 27-28.	12	72.08	26	14.45	85.43	85.38	170.81	160.00	93	2.92	17.2	2953	2375	270	89	11

MAY 11, 1887.

By FREDERICK A. SCHEFFLER, M. E., Erie, Pa.

CHICAGO, ILLS.

North Side Pumping Station, October, 1887.

PRELIMINARY REPORT OF

PROF. ROBERT H. THURSTON, Director Sibley College, Cornell University, Ithaca, N. Y.; J. S. COON, C. E., Burdett, N. Y.; JAS. N. WARRINGTON, C. E., Chicago, Ills.

Department of Public Works,
Commissioner's Office, October 16, 1887.

Messrs. Thurston, Coon & Warrington, *Experts to Test the "Gaskill Pumping Engines," North Side Pumping Station :*

Gentlemen—In conducting the test of the machinery, we desire that you determine both the capacity and duty of the machinery.

In the performance of said duty and capacity, the steam pressure shall not exceed eighty (80) lbs. per square inch in the boilers, and the piston speed of the engines shall not exceed one hundred and twenty (120) feet per minute.

The coal used shall be ordinary Anthracite Coal, and *the duty shall be performed without any deduction from the coal consumed, of clinkers, or any friction of machinery, friction of water in pumps, or connections for the use of steam, for feed pumps, or any purpose due to the operation of the machinery as a whole.*

Also, to determine the capacity of said engines, boilers, pumps and appendages, in U. S. gallons, per day of twenty-four hours, under a maximum head of one hundred and fifty (150) feet above the surface of water in suction-well, with a piston speed of one hundred and twenty (120) feet per minute.

One or both the engines may be tested at one time, at the option of the experts.

Your committee will also determine and report if the design, material and workmanship, either as a whole or in part, are excellent, ample and sufficient for the purpose intended and comply with the specifications for the same, a copy of which is appended hereto.

Respectfully,

GEO. B. SWIFT,
Commissioner of Public Works.

CHICAGO, October 29, 1887.

HON. GEO. B. SWIFT, *Commissioner of Public Works* :

SIR—The undersigned, the committee appointed by you to make a trial of the new Gaskill pumping engines, now in operation at the North Side pumping station, beg leave to submit the following report of the results so obtained :

A trial of each set of machinery was made, beginning on Monday, the 24th inst., with the west pair of engines, the trials being of twenty-four hours' duration each, that period having been adjudged by the committee sufficient to settle the questions referred to it by the city. A supplementary trial was also made of the engines first tested to determine the efficiency of the boilers and the distribution of heat in the engines. The data obtained and the calculated results will be submitted in full detail later.

The conclusions of your committee are as follows :

The contract made with the Holly Manufacturing Co., of Lockport, N. Y., the builders of the engines, binds them to furnish two sets of pumping machinery, each capable of delivering **12,000,000 gallons** of water per day of twenty-four (24) hours, against a head of one hundred and fifty (150) feet, the piston speed not exceeding one hundred and twenty (120) feet per minute, and to develop a duty of not less than **95,000,000 foot-pounds** per one hundred (100) pounds of coal burned in the boilers, no allowance for ashes or any other wastes being made.

The contractors are also called upon to furnish engines of good design, construction and material.

Your committee find, upon test, that the engines furnished by these contractors are of good design, of excellent workmanship and, so far as they have been able to judge by inspection, of the best of material. This committee is also able to certify that the stipulated capacity has been secured, and that the duty actually obtained is considerable above that demanded, **exceeding one hundred million (100,000,000) foot-pounds** of work done per one hundred (100) pounds of coal consumed.

Very respectfully,

R. H. THURSTON,
J. S. COON,
JAS. N. WARRINGTON,
Committee.

FINAL REPORT OF TRIALS FOR DUTY AND CAPACITY OF GASKILL PUMPING ENGINES

AT NORTH SIDE WATER WORKS, CHICAGO, ILLS., OCTOBER 24–28, 1887.

HON. GEORGE B. SWIFT, *Commissioner of Public Works, Chicago, Ills.:*

SIR—Your committee herewith respectfully submit their final report on duty and capacity trials of the two sets of Gaskill pumping engines supplied to your city in 1887, by the Holly Manufacturing Company, of Lockport, N. Y.

DESCRIPTION OF PUMPING ENGINES.

The machinery tested is that contained in the recent addition to the North Side Water Works, and comprises two independent sets of pumping engines, each set being really two independent and complete engines coupled to a common shaft, with cranks at right angles.

The engines are of the compound, condensing, horizontal beam and fly-wheel type. Each half of each engine has two steam cylinders situated in a vertical plane, and parallel to each other, the high-pressure being above the low-pressure, and very close together, making the steam passages between them as short as possible and keeping the clearance spaces down to a minimum.

The pistons being connected by short links to the short vertical beam, at opposite ends, travel in opposite directions.

The steam cylinders, sides and heads, are steam jacketed.

The double-acting pump plungers are in line with either low-pressure steam cylinder, and connected direct.

The high-pressure cylinder transmits its power to the pump plunger through the beam, which latter is between the steam cylinders and pumps.

The upper end of the beam is also connected with the fly-wheel shaft, the pedestal of which rests on top of the pump delivery chamber.

The low-pressure steam inlet valves, which are also the high-pressure exhaust valves, and the low-pressure exhaust valves are multiported, or "gridiron" slides worked by eccentrics from a lay-shaft lying alongside the high-pressure cylinder. The high-pressure inlet valves are double-beat poppets, operated by a "drop" cut-off, adjustable by hand.

The valve gear gives a very good steam distribution, favorable to economy in the consumption of coal.

The pump valves are numerous and very small, having a lift of only five-sixteenths inch, and a clear water way of about one and one-fourth square inch. The valves, faced with rubber, are noiseless and very efficient, as is shown by the small loss of action, or slip, being about 2¾ per cent. on the west engine, and about 3.2 per cent. on the east engine.

All bearing surfaces are ample for long-continued and uninterrupted service, while the design and construction are to be commended for great strength throughout. The workmanship is also excellent.

PRINCIPAL DIMENSIONS OF THE ENGINES.

Diameter of high-pressure pistons	27 inches.
Diameter of low-pressure pistons	54 inches.
Stroke of pistons and plungers, nominal	40 inches.
Diameter of high-pressure piston-rods	3.5 inches.
Diameter of low-pressure piston-rods	4.5 inches.

CLEARANCE, IN TERMS OF CYLINDER VOLUMES.

High-pressure, back end	3.72 per cent.
High-pressure, piston rod end	3.77 per cent.
Low-pressure, back end	2.102 per cent.
Low-pressure, piston rod end	2.055 per cent.
Diameter of plungers	30 inches.
Diameter of plungers, rods	5 inches.

STROKE OF PUMP PLUNGERS OBSERVED.

East engine, east plunger	39.687 inches.
East engine, west plunger	39.781 inches.
West engine, east plunger	39.812 inches.
West engine, west plunger	39.812 inches.

BOILERS.

Steam is supplied to the engines from boilers of the horizontal, return tubular type, in brick setting. They were also furnished by The Holly Manufacturing Company. The steam pipe passing from the boilers, passes through the up-take for a considerable distance, thus exposing the steam to the superheating action of the escaping gases. The superheating actually realized is slight, but it ensues dry steam.

DIMENSIONS OF BOILERS.

Diameter of shell	66 inches.
Length of shell	18 feet.
Number of tubes	66
Diameter of tubes	4 inches.
Diameter of steam drum	33 inches.
Width of grate	5 feet.
Length of grate	5.25 feet.
Grate area	26.25 sq. feet.
Area of chimney	44.4 sq. feet.
Height of chimney	125 feet.
Length of flues connecting to chimney	16 feet.
Area of flues connecting to chimney	22. sq. feet.
Heating surface, water	1438.7 sq. feet.
Superheating surface, steam pipe	182. sq. feet.
Draught area, through tubes	5.76 sq. feet.
Ratio grate to water heating surface	1 to 54.8

Feed water for the boilers is supplied by feed pumps driven by the pumping engines. The feed water is taken from the hot well, from which it passes through a small heater whose source of heat is the condensations from the jackets. The jacket condensations are also pumped into the boiler. By this means an average temperature of 180° was imparted to the feed water.

CONTRACT REQUIREMENTS.

The contract with the Holly Company binds them to furnish two sets of pumping engines, capable of a combined delivery of **twenty-four million gallons** of water per day of 24 hours, against a head of one hundred and fifty (150) feet, at a piston speed not exceeding one hundred and twenty (120) feet per minute, at a boiler pressure not exceeding eighty (80) pounds per square inch, and to develop a duty of not less than **ninety-five million foot-pounds (95,000,000) per one hundred (100) pounds of coal burned at the boilers, no allowance for ashes, friction or any wastes to be allowed.** It was further stipulated that the above quantity of water should be delivered when two-thirds of the boilers furnished were supplying steam to the engines, and that **the duty should be based on the water delivered, as determined by weir measurement.**

MANNER OF CONDUCTING THE TRIALS.

Local conditions rendered it inexpedient to run both engines at once, during the trials. It was decided to test each engine separately, for a period of 24 hours.

Since a boiler test could not be made in conjunction with a duty and capacity test without lowering the temperature of the feed water (in transit through weighing tanks), each engine was tested in exactly the conditions which maintain in every-day service.

Very satisfactory facilities for gauging the water delivered by the pumping engines had been provided. A weir flume of ample length having a cross-section 11 feet wide by seven feet deep, was provided at its entering end with gratings to insure an even flow of water in the flume, and a weir notch with carefully prepared metallic knife edges at its exit end. A vernier hook gauge, located in a box at one side of the flume, indicated the height of water on the weir notch. All precautions, as recommended by Mr. James B. Francis, were observed in connection with the weir measurements, and Mr. Francis' formulas were used in the computations.

In ordinary service the water pumped by the engines is delivered directly into the city mains. During the tests, communication with the city mains was shut off, and the water delivered to the measuring flume, from which it passed over the weir notch, to a launder emptying into Lake Michigan.

Since the measuring flume was below the engines, in order to create an artificial head for the engines to pump against, a gate value in the delivery main, near the engine-house, was partially closed, until a throttling resistance corresponding to a total head of about 150 feet was obtained.

In the beginning of the first trial nearly an hour elapsed before the resistance and engine speed could be adjusted conformably to the conditions of the contract. Hence, the capacity of the (west) engine should be based on the time during which it was running under normal pressure and speed. The duty, of course, is figured for the whole time. The (west) engine was run, however, more than 24 hours, under normal conditions.

Of the six boilers in the battery, two, Nos. 1 and 2, were used throughout the tests, and these had been under steam for several days prior to the trials.

In consequence of the peculiar conditions under which the engines were compelled to work, it was decided best to make a running start.

The first trial was with the west engine, October 24, 1887, and was conducted as follows: With the engine running, the fires under boilers Nos. 1 and 2 were quickly hauled, and the grates cleaned, the steam for keeping the engine running coming mainly from boiler No. 3, which was under steam. Kindling wood was quickly placed under boilers 1 and 2, the wood being charged to the experiment as equivalent to four-tenths (0.4) its weight in coal. At the instant the fire was lighted boiler No. 3 was shut off, time was taken, 2.29 P. M., and the test began. The engine counter was taken, steam pressure and height of water in the boilers carefully noted. At the close of the trial, after the last firing, when the steam pressure had fallen to the point at which it stood when the fires were lighted, the water level in the boilers being also the same as at beginning of trial. The trial was closed at 3 59 P. M., October 25. The fires were immediately hauled, with the intention of picking out any unburned coal, but not enough remained unburned to make any account of. The steam pressure in boiler No. 3 was kept below that of the boilers supplying steam to the engine.

The trial with the east engine began October 26th, and was conducted as follows: With the engine running, the fires under boilers No. 1 and 2, supplying steam to the engine, were allowed to burn low, and when it was decided to commence cleaning the fires, time was taken, 11.35 A. M., and the test began. The engine counter was taken, steam pressure and water level in the boilers recorded. The fires were then cleaned, all coal added being charged to the experiment till the close of the trial, when, after the last firing, the fires were brought, as nearly as could be judged, to the same condition as at the beginning, the steam pressure and water level in the boilers being also the same. When it was decided that the fires needed cleaning, time was taken, 11.40 A. M., October 27th, the engine counter taken, and the trial declared closed.

It was predicted that the east engine would show the higher duty, and the tests verified this.

The coal was weighed in iron barrows, on a small platform scales, 300 pounds net, at each barrowful. During the trials the scales were sealed by the local Sealer of Weights and Measures.

The coal used was clean "egg" anthracite, such as is supplied to the city water works.

Two observers were in constant attendance in the boiler-room to look after the coal

Two observers were also in attendance at the weir, where observations on the hook gauge were made every ten minutes.

A metallic float gauge, inclosed in a gauge box in the pump well, indicated the water level in the latter. The zero of this gauge was carefully referred to the center of the pressure gauge in the force main.

The steam temperature was indicated by a thermometer screwed into the steam pipe near the engine.

The force main pressure gauge, and the thermometers, were prepared specially for these tests.

Observations were made, as to the quality of the steam, with a barrel calorimeter. These observations indicated dry steam. The thermometer in the steam pipe showed about 4 degrees super-heat.

Observations were made every fifteen minutes on the engine counter, pressure gauges and thermometers in the engine and boiler rooms.

A few sets of indicator diagrams were taken, to obtain an idea of the steam distribution, but more particularly to ascertain the friction of the engine and percentage of useful effect. The friction was thus ascertained to be 7.64 per cent., and the efficiency 92.36 per cent.

It is to be borne in mind that the duty, as required by the contract, and as given in the tables, is figured on the weir measurement. If figured on plunger displacement, as is customary, the mean duty for both engines would have been about **103,072,000 foot pounds.**

The necessary data for determining the steam used per indicated horse-power, was obtained during the boiler trial, to be described hereafter.

The tabulated results for the second trial of the west engine which was run during the boiler trial, are merely incidental. The drop in the duty during the second trial is due solely to the lowering of the temperature of the feed water from 181 degrees to 148 degrees. Making the proper correction for this loss of temperature (caused by the feed water standing in the measuring tanks) the west engine, for the boiler trial, should be credited with a duty of **100,277,000 foot pounds.**

TABULATED RESULTS.

	WEST ENGINE	WEST ENGINE.	EAST ENGINE.	WEST ENGINE.
	Neglecting 1st 57 Min.	Whole Time.		Boiler Trial.
Duration of trial (hours)	24.55	25.50	24.08	12 283
Revolutions, total.	27,505.	28,480.	27,084	13,682.
Revolutions per hour.	1,120 37	1,116.86	1,124 59	1,113.90
Revolutions per minute	18.673	18 614	18.743	18.565
Piston speed in feet, per minute—				
Actual	123.90	123.51	123.46	123.19
For 12,000,000 gals. per day	118 2	......	118.3	......
Displacement, gals. per revolution	480.5	......	478.6	......
Average head at weir, feet	......	0.8191	0.8189	0.8162
Velocity of approach, feet per second	......	0.2411	0.2411	0.2411
Head at weir, corrected for velocity of approach, feet	......	0.820	0 81978	0.81708
Water pumped, cubic feet per second.	......	19.378	19.368	19.274
Water pumped, U. S. gals., per hour.	524,028.	521,810.	521,543.	519,008.
Water pumped, U. S. gals., per day	12,576,752.	12,523,464.	12,517,032	12,456.192.
Water pumped, U. S. gals.. total	12,864,882.	13,306,181.	12,560,494.	6,374,975.
Water pumped, U. S. gals., per revolution	467.74	467.21	464.83	465.93
Loss of action, or "slip," per cent.	2.659	2.766	3.105	3.029
Mean force, main pressure, pounds	......	59.843	59 397	60 372
Mean force, main pressure, feet head.	......	138.237	137.21	139.460
Mean height from gauge to water in pump well	......	13.649	14 072	14.013
Mean total head against pump, feet	......	151.886	151.280	153.470
Pounds of water raised against head.	......	110,986,856.	104,767,080.	53,173,666.
Foot pounds of work done	......	16,857,349,610.	15,849.163,862	8,160,562,521.
Fuel consumed in Coal equivalent—				
300 lbs. kindling wood, at 0.4 lbs.	......	120.	......	......
600 lbs. Indiana coal, at 0.7 lbs.	......	420.	......	......
Anthracite coal, "Egg"	......	16,500.	15,450	8,400.
Total fuel in Anthracite equivalent	......	17,040.	15,450	8,400.
Duty in pounds of water raised one foot per 100 lbs. coal	......	98.928,108.	**102,583,585.**	97,149,554.
AVERAGES—GENERAL.				
Steam temperature at throttle, deg. F.	......	Thermometer deranged	325 99	327.98
Steam pressure at throttle, by gauge	......	79.88	79.09	79.96
Steam pressure at throttle, absolute	......	......	93.58	94.45
Steam, degrees of super-heat	......	......	3.18	4.51
Steam pressure in low pressure jackets	......	61 65	Not Taken.	62.61
Feed water temperature at hot well	......	98.65	93.74	104.68
Feed water temperature before passing heater	......	115.25	110.54	123.86
Feed water temperature after passing heater	......	181.21	180 79	161.08
Feed water temperature as it entered boiler	......	181.21	180.79	148.04
Injection temperature	......	52.81	Not Taken.	51.36
Vacuum, inches of mercury	......	26.30	26 40	25.51
Steam pressure at boilers	......	80.11	80 01	80.02
Engine room temperature	......	74.7	75 57	78.68
External air temperature.	......	32.36	38 58	51.92
Ashes and clinker, total lbs	......	1,267.	1,202.5	852.
Combustible, lbs	......	15,773.	14,247.5	848.
Per cent. of ashes in coal	......	7.43	7.78	6.57

In the following table column (1) gives the indicated horse-power by five sets of indicator cards taken from the steam cylinders of the west engine. Column (2) gives the simultaneous horse-power in water pumped, as per weir discharge. Column (3) gives the horse-power by the plunger travel against the total head. Columns (4) and (5) give the per cent. of indicated horse-power utilized, being the ratios of columns (2) and (3) to column (1).

The difference between the averages of columns (1) and (3), viz.: 28.5 horse-power, is the friction of the engine, which, at that speed, would remain practically constant for varying heads, so that for any other head the duty of the engine can readily be calculated. Thus, for a head of 300. foot the duty would be increased 3.6 per cent. But this is only approximately correct, as under increased head increased steam pressure and cylinders would be necessary.

WEST ENGINE.

INDICATED AND EFFECTIVE HORSE-POWER.

HORSE-POWER.			PER CENT. OF I. H. P. UTILIZED.	
Indicated.	Weir Discharge.	Plunger Travel.	Weir Discharge.	Plunger Travel.
373.36	335.96	345.58	89.99	92.56
377.25	340.80	348.06	90.34	92.26
375.93	338.05	346.48	89.92	92.17
369.90	333.04	341.97	90.04	92.45
369.19	332.77	341.08	90.14	92.39
Averages...373.13	336.12	344.63	90.08	92.36

BOILER TRIAL.

The test of boilers Nos. 1 and 2 was conducted in precisely the same manner as with the trial of the east engine, and, in addition, the feed water was weighed. With the (west) engine running, the fires under boilers Nos. 1 and 2 were allowed to burn low, and when it was decided to commence cleaning fires time was taken, 10:23:30 A. M., October 28th, and the test began, the steam pressure and water level in the boilers being carefully noted. The fires were then cleaned, all coal added being charged to the experiment, until after the last firing. The fires were brought as nearly as possible to the same condition as at the beginning, the steam pressure and water level being also the same. When it was decided the fires needed cleaning time was taken, 10:40:30 P. M., October 28th, and the trial closed.

Incidentally, all observations in the engine room and at the weir were made during this test as during the others.

Excellent facilities were provided for weighing the feed water. The feed water was pumped (by the feed pump, driven by the main engine) into an elevated tank, No. 1. As needed, the feed was drawn from tank No. 1 into a weighing tank, resting on platform scales, each charge weighing about 2,000 pounds net. From the weighing tank the water passed into tank No. 3, and from the latter pumped into the boilers by an independent steam pump supplied with steam from boiler No. 6. The pressure in boiler No. 6 was kept below that in boilers No. 1 and 2.

In order that the resistance against the pumping engine should be the same as if it were pumping the feed water directly into the boilers, as during the other trials, a throttling valve was placed in the feed pipe between the pumping engine and tank No. 1, and adjusted so that the resistance against the pump equaled the boiler pressure. (See column 19 of the engine room log for October 28th.)

Columns 25 and 26 of boiler room log for October 28th, give the overflow from tank No. 1, as measured by a meter. This overflow is the water pumped, by the feed pump at the engine, in excess of the boiler requirements.

RESULTS OF BOILER TEST.

Duration of trial, hours	12.283

AVERAGE PRESSURES.

Steam pressure in boilers, gauge, lbs	80.02
Steam pressure in boilers, absolute	94.51
Atmospheric pressure, barometer, inches, mercury	29.50
Draught, force of, inches of water	.652

AVERAGE TEMPERATURES.

Feed water, degrees F	148.72
Steam, at engine throttle, F	327.98
Fire room	66.7
External air	51.9

FUEL.

Anthracite coal consumed (assumed to be dry), lbs	8,400.
Ashes and clinker from grates and ash-pits, lbs	552.
Per cent. of ash in coal	6.57
Combustible, lbs	7,848.
Coal consumed per hour	683.8
Combustible consumed per hour	638.9

FEED WATER.

Water pumped into boilers and apparently evaporated, lbs	82,774.5
Equivalent evaporation into dry steam from and at 212° F., lbs	91,335.4
Equivalent British thermal units derived from fuel	82,202,600.

ECONOMIC EVAPORATION.

Water evaporated, per pound of coal, at observed temperatures and pressures, lbs	9.854
Equivalent evaporation, per pound of coal, when working under normal conditions, viz. : with feed water at 181° instead of 148.7°, lbs	10.169
Equivalent evaporation per pound of coal, from and at 212° F., lbs	10.873
Equivalent evaporation, per pound of combustible, lbs	11.638

COMMERCIAL EVAPORATION

Equivalent water evaporated, per pound of coal, with one-sixth refuse, from feed water at 100° F., into steam at 70 pounds, gauge, lbs	8.436

RATE OF COMBUSTION.

Coal burned per hour per square foot of grate, lbs.	13.026
Coal burned per hour, assumed with one-sixth ash, per square foot of grate, lbs.	14.604
Coal burned per hour, assumed with one-sixth ash, per square foot of water heating surface, lbs.	0.2664
Coal burned per hour, assumed with one-sixth ash, per square foot of least area for draught, lbs.	133.11

RATE OF EVAPORATION.

Water evaporated per hour, from and at 212° F., per square foot of heating surface, lbs.	2.584
Water evaporated per hour, from feed at 100° F., into steam at 70 pounds, gauge, per square foot of grate, lbs.	123.2
Water evaporated per hour, from feed at 100° F., into steam at 70 pounds, gauge, per square foot of heating surface, lbs.	2.247
Water evaporated per hour, from feed at 100° F., into steam at 70 pounds, gauge, per square foot of least area for draught, lbs.	1,122.9

COMMERCIAL HORSE-POWER. (ONE.)

On basis of 30 pounds water per horse-power, per hour, evaporated from feed at 100° F, into steam at 70 pounds, gauge, horse-power.	107.76

COAL AND WATER PER HORSE-POWER PER HOUR.

Coal burned during boiler trial, lbs.	8,400.
Coal which would have been burned if feed water had been 181° instead of 148.7°, lbs.	8,139.5
Effective horse-power of west engine for whole time of boiler trial, figured on weir measurement	335.54
Net horse-power of same, by plunger travel	344.04
Indicated horse-power of same	372.5
Coal burned per hour (8,139.5÷12.283), lbs.	662.66
Coal burned per hour, per indicated horse-power	1.779
Coal burned per hour, per net horse-power	1.926
Water per hour, lbs.	6,738.95
Water per hour, per indicated horse-power, lbs.	18.091
Water per hour, per net horse-power, lbs.	19.58

Your committee take pleasure in stating that the performance of the engines and boilers is, in their opinion, eminently satisfactory, both the duty and capacity being in excess of the guarantee; the mean duty for both engines being over 6 per cent. in excess of the requirements of the contract.

In conclusion, we beg leave to express our thanks to the contractor, to officials at the City Hall, and particularly to Mr. T. T. Johnston, of the City Engineer's Office, for courtesies and valuable assistance.

Very respectfully,

ROBERT H. THURSTON,
J. S. COON,
JAS. N. WARRINGTON.

PHILADELPHIA, PA.

NOVEMBER, 1887.

PRELIMINARY REPORT OF JOHN E. CODMAN, C. E.

PHILADELPHIA, December 16, 1887.

MR. JOHN L. OGDEN, *Chief Engineer Water Bureau:*

SIR—The undersigned, expert appointed by you to make the capacity and duty tests of the new 20,000,000 gallon Gaskill pumping engine now in operation at the Spring Garden Pumping Station, respectfully submits the following preliminary report of the results obtained:

The data and calculated results will be submitted in full details later.

The contract made with the Holly Manufacturing Company, the builders of the engine, binds them to furnish a pumping engine capable of delivering **20,000,000 gallons** of water per diem of 24 hours against a head of 165.5 feet above water in the dump well, speed of piston not to exceed 140 feet per minute, and to develop a duty of **110,000,00 foot-pounds** per ten (10) pounds of water delivered to boiler into steam of 100 pounds gauge pressure.

During the trial test of 24 hours we found the engine to run smoothly without heating or vibration, and, so far as we are able to judge, the material and workmanship are of the best.

We also find that the engine has pumped the stipulated quantity of water, and at the same time the duty actually obtained is **125,022,730 foot-pounds**, exceeding the contract by over **15,000,000 foot-pounds**, or nearly **14 per cent.**

I also find that the duty of the combined plant of boilers and engine together, calculated from the actual of coal consumed, to be **121,854,515 foot-pounds.**

Very respectfully,

JOHN E. CODMAN,

Expert for Philadelphia Water Bureau.

PHILADELPHIA, PA.

Spring Garden Pumping Station, November, 1887.

FINAL REPORT OF JOHN E. CODMAN, C. E.

Philadelphia, January 10, 1888.

Mr. John L. Ogden, *Chief Engineer Bureau of Water :*

Sir—The following report on the duty and capacity tests of the 20,000,000 gallon Gaskill Engine recently completed at the Spring Garden Pumping Station, is respectfully presented :

By the requirements of the contract, a capacity and a duty test were to be made—the latter within two months after the engine shall have been put in operation. By mutual agreement both tests were made at the same time.

The duty required by the specifications was **one hundred millions (100,000,000) of foot-pounds**, with a consumption of one hundred (100) pounds of coal on the basis of ten (10) pounds of water evaporated per pound of coal; the duty to be computed by the following formula :

$$\frac{P \times N \times H \times 100}{W} = \text{Duty}$$

in which P = pounds of water delivered per stroke, N = the number of strokes during trial, H = total fluid resistance, including static and frictional heads measured to the surface of water in pump well, allowing one pound for friction through the pump and passages, and W = number of pounds of coal on the basis of ten pounds of water evaporated per pound of coal.

The duty guaranteed by the contractors was **one hundred and ten millions (110,000,000) foot-pounds.**

The trial of 24 hours duration began at 8 A. M., Tuesday, November 29th, and terminated at 8 A. M., Wednesday, November 30, 1887.

METHOD OF CONDUCTING TEST.

It was agreed to take half-hourly observations from all the attachments on both boilers and engine. For this purpose one steam gauge on the engine was tested for reading steam pressure thereon; one steam gauge out of the five inspected was selected and tested for reading steam pressure on the boilers; one pressure gauge on the engine was tested for reading the water pressure; and the distance from center of gauge to surface of water in forebay measured.

The coal consumed was weighed in the small cars used at the station, and after inspection was run into the boiler room. The firing was done by the regular men.

The water for the boilers was pumped into a wooden tank capable of holding about 2,700 pounds, weighed, and run thence into an iron tank of about 9,000 pounds capacity, from which an attachment was made to the pumps feeding the boilers. Both tanks were set on platform scales, which had been inspected and corrected.

In the agreement a clause, voluntarily made by the contractors, stipulated that the engine while on test should pump all feed water into boilers, and no allowance be made for steam so used. For this purpose four plunger feed pumps were connected to a rocker arm attached to a trunnion on the walking-beam center. Two of these were regulated to pump water continuously into the boilers. In calculating the duty no allowance has been made for this quantity of work, aggregating, during the 24 hours, 66,000,000 foot-pounds. In making a comparison of duty tests with other engines it is only fair to state that this amount of work is commonly done by an extra donkey pump supplied with steam from another set of boilers kept running for this purpose, thus involving not only extra labor but also coal and boiler capacity.

It was agreed to check the water observations in the large tank at the end of each hour. For this purpose a gauge reading in inches from the bottom of the tank was so placed that the height of water could be plainly seen. At the beginning of the test the height of water was read on the gauge, and at the end of each hour the water was brought to the same point, and the quantity for the hour weighed out of the small tank and checked off by the observers. The specifications required that ten pounds of water should be considered as a pound of combustible, therefore particular attention was given to the accuracy of the observations on the amount of water weighed out to the boiler.

The high and low-pressure cylinders are both steam-jacketed with live steam from the boilers. In the ordinary running of the engines, the condensed water resulting therefrom is carried through a coil heater and steam trap and pumped back into the boilers, increasing the temperature of the feed water to nearly 212° F.

On the trial this jacket water was cut out from the feed pumps and a steam trap so placed that the discharge from it could be collected in a barrel placed upon platform scales, weighed and checked every hour. In calculating the duty this jacket water has not been deducted from the amount actually pumped into the boilers and charged to the contractors, nor was it permitted to be *trapped* back into the boilers without being accounted for.

The specifications requiring the duty to be calculated from ten pounds of water equivalent to one pound of coal, made the engine and boilers two separate plants. The management of the boiler fires and all the attachments in the boiler-room were under the direction of the Bureau of Water, but all observations were taken jointly.

The five boilers used in furnishing steam were made from designs and drawings prepared in the Bureau, and were calculated to carry 100 pounds of steam pressure and to furnish a sufficient supply of steam to enable the engine to fulfill the contract. On a preliminary trial it was found that four boilers would furnish ample steam for the engine, but it was considered best to use the five in order to maintain a constant steam pressure in cleaning fires.

It was agreed to clean the fires four times during the twenty-four hours and weigh all ashes and clinkers dry. In order to bring the fires to the same condition when the trial ceased as they were in the beginning, directions were given to clean the fires in regular succession three hours before the trial began, cleaning two furnaces at a time in the several boilers at intervals of one-half hour each, and covering a period of two hours, and securing clean fires one hour before the trial began—the same order following in the successive four cleanings, and beginning the last cleaning three hours before the end of the trial, thus leaving the fires as nearly as possible in the same condition as when the trial started.

For obtaining the temperature of the escaping gases, a pyrometer was placed in the flue back of the boilers. The accuracy of the pyrometer was tested by taking a calorimetrical observation with a known weight of iron and a given quantity of water of known temperature and found to be correct within about 20° F. An attachment was made to the steam pipe to take calorimetrical observations on the quality of the steam.

The coal used was rather an inferior quality of Schuylkill nut, containing about 25 per cent. of clinker and ash, besides being very wet. A sample of 370 pounds was taken and dried upon the boiler flue for twenty-four hours and found to contain 6 per cent. of moisture—the evaporation of which is credited to the boilers and is shown in the tabulated report of the results of the boiler trials.

The management of the engine during the trial was under the direction of the contractors exclusively. Two indicators were placed on the high pressure cylinders and two on the low, and indicator cards from both steam cylinders were taken simultaneously at intervals during the twenty-four hours. One of these cards has been enlarged and plotted to scale, and from it has been determined the quantity of steam used per hour, quantity of steam per horse-power per hour, together with the volume of steam admitted to cylinders. The isothermal curve and the adiabatic curve are both traced on the diagram, and the area of the actual card in per cent. of the theoretical is given.

The specifications, section 12, make it optional with the Chief Engineer of the Bureau of Water, whether the capacity shall be determined by weir measurement or by plunger displacement. There being no pumping main nor weir available for that method of measurement, by your direction the capacity was determined by plunger displacement, and the duty and the capacity test were run at the same time. By partially closing an outside stop-valve the water gauge was made to indicate the required pressure, including static and frictional head.

CALCULATED DATA.

Volume of high-pressure cylinder, including clearance in cubic feet	24.128
Volume of low-pressure cylinder, including clearance in cubic feet	96.80
Total cubic feet in both cylinders, including clearance, cubic feet	120.93
Ratio of volume of low-pressure cylinder to volume of high-pressure cylinder,	4.01
Cubic feet in one volume calculated from indicator card	7.56
Ratio of expansion by pressure	12.44
Ratio of expansion by volume	12.80
Area of theoretical card, square inches	65.45
Area of actual card, square inches	53.16
Area of actual in per cent. of theoretical	81.22
Indicated horse-power	772.7
Pounds of steam per stroke	1.95
Pounds of steam per hour by card, including jacket	9,548.1
Pounds of steam per indicated horse-power per hour by card, including jacket	12.36
Pounds of steam per indicated horse-power per hour, from water actually weighed into boilers	14.94
Per cent. of water accounted for by indicator card and jacket water	82.
Total amount of jacket water by actual weight during the 24 hours	26,276.
Jacket water per hour	1,095

Eleven calorimetrical observations on the quality of steam were made during the test at intervals of about two hours each.

Weight of water heated, pounds	200.
Weight of steam condensed, pounds	10.06
Initial temperature, deg. F.	57.39
Final temperature, deg. F.	111.14
Elevation of heat of water by condensation of steam, deg. F.	53.75
Absolute steam pressure, pounds	111.35
Latent heat units at observed pressure	878.95
Latent heat units of steam as found	848.974
Quality of steam	0.965
Percentage of water contained in steam	3.5

AVERAGES OF OBSERVED DATA.

Averages of the observed data during the 24 hours—complete table of which it is not necessary to give—are as follows:

Steam pressures, November, 29th, 8 A. M.:

Boilers	Nos. 34.	35.	36.	37.	38.
Pressure by gauge	98	98	98	98	98
November 30th, 8 A. M.					
Pressure by gauge	97	97	97	97	97
Average for 24 hours, 49 readings	96.53	96.53	96.53	96.53	96.53

WATER GAUGES—BOILERS.

Levels in inches of water in glass tube, 8 A. M., November 29th:

Boilers	Nos. 34.	35.	36.	37.	38.
Levels, inches	8	7½	8½	8	7¼=39¼
Levels in inches of water in glass tube 8 A. M., November 30th	Nos. 34.	35.	36.	37.	38
Levels, inches	7¾	7¼	8¼	8	7¾=39

TEMPERATURE OF FEED WATER.

The temperature of the feed water in the large tank was taken three times during each hour, but the distance from the feed pumps to the boilers was so great that a thermometer was placed in the feed pipe near the boilers, from which the temperature of the feed water entering boilers was taken:

	Nov. 29th, 8 A. M.	Nov. 30th, 8 A. M.	Average of 49 readings.
In tank	87	97	98.2
In pipe before entering boilers	86	96	96.3

TEMPERATURE OF ESCAPING GASES.

	Nov. 29th, 8 A. M.	Nov. 30th, 8 A. M.	Average of 49 readings.
Pyrometer readings	612° F.	650° F.	555° F.

TEMPERATURES.

Fire room, 8 A. M., November 29th	66° F.
Fire room, 8 A. M., November 30th	64° F.
Fire room, average of 49 readings	66° F.
Outside air, 8 A. M., November 29th	36° F.
Outside air, 8 A. M., November 30th	26° F.
Outside air, average of 49 readings	35° F.
Mean barometer readings, 24 hours, from Signal Service Bureau	30.655
Mean pressure, pounds	15.05

ANEMOMETER READINGS.

Velocity of air passing through fire grate per minute:

Average of 4 readings:

Boiler—No. 34	235 feet.
No. 35	237 feet.
No. 36	220 feet.
No. 37	260 feet.
No. 38	225 feet.
Average of 20 readings	235 feet.

COAL TO BOILERS.

Total pounds of coal weighed	44,014
Less 6 per cent. for moisture	2,641
Dry coal, pounds	41,373
Less 26.38 per cent. of waste	10,088
Total combustible, pounds	31,285
Evaporation of boilers per pound of combustible from and at 212° F., pounds	10.35

The following order was observed in cleaning fires:

Boilers 34 and 36, furnaces Nos. 1 and 2. Time 5 A. M., 12 noon, 5 P. M., 12 M., 5 A. M.

Boilers 34 and 37, furnaces Nos. 2 and 1. Time, 5.30 A. M., 12.30 P. M., 5.30 P. M., 12.30 A. M., 5.30 A. M.

Boilers 35 and 37, furnaces Nos. 1 and 2. Time, 6 A. M., 1 P. M., 6 P. M., 1 A. M., 6 A. M.

Boilers 35 and 38, furnaces Nos. 2 and 1. Time 6.30 A. M., 1.30 P. M., 6.30 P. M., 1.30 A. M., 5.30 A. M.

Boilers 36 and 38, furnaces Nos. 1 and 2. Time, 7 A. M., 2 P. M., 7 P. M., 2 A. M., 7 A. M.

Ash, clinker and unburnt coal were weighed back as follows:

November 29th, 2.15 P. M	2,850 pounds
November 29th, 7.10 P. M	2,265 pounds
November 30th, 2.15 A. M	3,139 pounds
November 30th, 7.15 A. M	1,834 pounds
Total	10,088 pounds

Percentage of non-combustible

$$\frac{10{,}088 \times 100}{41{,}373.16} = 24.38$$

The records in the engine room consisted of readings from the engine counter, steam gauge on steam pipe, water pressure gauge connected with the pumping main, and the vacuum gauge every half hour, with the following initial and final readings and averages:

Readings at 8 A. M., Nov. 29th:

Steam gauge.	Water gauge.	Vacuum.
96	71.5	27

Readings at 8 A. M., Nov. 30th:

Steam gauge.	Water gauge.	Vacuum.
95	73.5	27.5

Average readings for 24 hours:

Steam gauge.	Water gauge.	Vacuum.
49 readings,	49 readings,	49 readings,
95.3	70.99	27.5

RESULTS FOR CAPACITY.

Counter on engine 8 A. M., Nov. 29th	719,001
Counter on engine 8 A. M., Nov. 30th	745,011
Revolutions of engine in 24 hours	26,010
Revolutions per hour	1,083.7
Revolutions per minute	18.063
Number of strokes in 24 hours	104,040
Diameter of plungers, feet	3.00
Stroke of plungers, feet	4.00

From data obtained on previous tests of the same arrangement of pump-valves and plungers, it was agreed to allow 2 per cent. for slip or loss of action due to supposed imperfect filling of the pump chambers.

Area of 3 feet, diameter of plunger	7.0686 sq. feet
Less one-half area of 6 inch, diameter of plunger rod	.0981 sq. feet
	6.9705 sq. feet

Cubic feet per stroke 6.9705 × 4=27.882 cubic feet.

Number of strokes in 24 hours=104,040.

Number of cubic feet in 24 hours=104,040 × 27.882=2,900,843.28

Number of gallons in 24 hours=2,900,843.28 × 7.48=21,698,307.7

Less 2 per cent. for slip	433,966.2
Total capacity	**21,264,341.5**

During this time the engine made 18.063 revolutions per minute, or a piston speed of 144.5 feet per minute.

The contractor's guarantee was a delivery of **20,000,000 gallons**, with a piston speed of 140 feet per minute, or 17½ revolutions per minute.

The capacity at the contractor's guarantee is as follows:

Number of strokes in 24 hours	100,800
Cubic feet per stroke	27.882
Gallons per cubic foot	7.48

Number of gallons in 24 hours=

27.882 × 7.48 × 110,800=**21,022,848**

Less 2 per cent. for slip	420,456
Total capacity	**20,602,392**

which is about 3 per cent in excess of guarantee.

Capacity during test	**21,264,341**

which is over 6 per cent. above the guarantee.

RESULTS FOR DUTY.

Engine counter at 8 A. M., Nov. 29th	719,001.
Engine counter at 8 A. M., Nov. 30th	745,011.
Revolutions in 24 hours	26,010.
Revolutions per hour	1,083.7
Revolutions per minute	18.063
Piston speed per minute during 24 hours, feet	144.5
Mean head by water pressure gauge, 49 readings, pounds	70.99
Mean head by water pressure gauge, feet	163.937
Mean distance from center of water pressure gauge to level of water in forebay, feet	25.00
Contract allowance for friction of water passages in pumps	2.3093
Total head in feet, as per printed specifications—H—	191.246
Temperature of water in forebay	49° F.
Weight of one cubic foot of water at that temperature	62.41
Cubic feet of water displaced at one stroke of plunger	27.882
Pounds of water at one stroke—P—	1,740.116
Number of strokes in 24 hours—N—	104,040.
Total weight of water weighed to boilers	286,982.
Less 3½ per cent. for entrained water in steam	10,044.4
	W—276,937.6

$$\text{Then } D = \frac{1740.116 \times 104{,}040 \times 191.246 \times 100}{27{,}693.76} =$$

125,022,730 foot pounds per 100 pounds of coal, or **15,022,730 in excess of guarantee**, or nearly 14 per cent.

The contractors claim that 2 per cent. for slip shall not be deducted from the duty obtained from the reading of the pressure gauge, for the reason that the pressure by gauge remains constant, and this difference, if any, is caused by the plunger forcing the water under the valves, and the work done by the pumps should be calculated from the pressure as given by the water gauge.

Deducting this 2 per cent. for slip, the duty becomes **122,522,276 foot pounds** per 100 pounds of coal or **12,522,276 foot pounds** in excess of guarantee—or nearly 12 per cent.

Very respectfully,

JOHN E. CODMAN,
Expert for Philadelphia Water Bureau.

OBSERVATIONS FOR 24 HOURS, TAKEN BY A. MARICHAL AND JAMES G. DAVIS. LOG OF BOILER TEST, AT SPRING GARDEN STATION, NOV. 29 AND 30, 1887. BUREAU OF WATER, PHILADELPHIA.

Time.	Pressures.			Temperatures.						Fuel.		Feed-Water.		Water in Jacket.
	Mean Barometer during test.	Steam Gauge.	Draught Gauge.	External Air.	Boiler Room.	Flue.	Feed-Water.		Steam.	Pounds of Coal.	Pounds of Ashes.	Time.	Pounds.	
							Pipe.	Tank.						
A. M.	Ins.	Lbs.	Ins.	Deg.	Deg.	Deg.	Deg.	Deg.	Deg.	Lbs.	Lbs.	A. M.		
8.00	30,655	98	.06–.065	36	66	612	86	87	Average Temperature of Steam for Twenty-four Hours.					
8.30		97	.06–.065	37	69	512	88	90						
9.00		97	.06–.065	38	69	480	91	91		2,500		9.00	12,925	1,440
9.30		93	.06–.065	38	65	565	88	92						
10.00		96	.06–.065	46	64	450	104	108		2,114		10.00	14,910	1,090
10.30		95	.06–.065	46	64	425	96	102						
11.00		96	.06–.065	52	64	456	96	98		1,030		11.00	9,699	1,122
11.30		96	.06–.065	52	64	485	96	100						
Noon. 12.00		98	.06–.065	52	65	565	98	100		1,818		Noon. 12.00 P. M.	11,409	1,084
P. M. 12.30		97	.06–.065	53	64	525	96	100						
1.00		97	.06–.065	53	64	540	97	99		2,000	2,850	1.00	11,177	1,078
1.30		98	.06–.065	57	65	490	96	99						
2.00		96	.06–.065	57	65	490	96	99		2,126		2.00	11,198	1,097
2.30		96	.06–.065	40	67	525	97	99						
3.00		97	.06–.065	39	67	525	98	98		1,500		3.00	11,087	1,057
3.30		97	.06–.065	38	66	515	97	98						
4.00		97	.06–.065	37	63	525	97	100		1,294		4.00	11,463	1,072
4.30		97	.06–.065	36	65	525	96	99						
5.00		96	.06–.065	34	66	460	98	98		1,500	2,265	5.00	12,721	1,058
5.30		95	.06–.065	34	64	500	95	96						
6.00		97	.06–.065	32	64	675	97	98		2,354		6.00	11,402	1,065
6.30		96	.06–.065	32	63	690	97	98						
7.00		94	.06–.065	31	62	625	97	100		1.500		7.00	11,904	1,075
7.30		95	.06–.065	31	62	640	97	99						
8.00		96	.06–.065	31	62	525	97	98		2,452		8.00	12,393	1,068
8.30		97	.06–.065	31	62	650	95	96						
9.00		98	.06–.065	30	60	560	96	97		1,650		9.00	12,635	1,089
9.30		97	.06–.065	30	60	500	97	98						

10.00	....	97	.06–.065	30	61	500	98	99	Average Temperature of Steam for Twenty-four Hours.	2,000	....	10.00	12,021	1,120
10.30	....	97	.06–.065	30	62	500	98	99		....	....	....	....	....
11.00	....	96	.06–.065	30	62	600	96	99		1,682	....	11.00	11,881	1,066
11.30	....	97	.06–.065	30	62	600	97	100		....	....	....	....	....
Midnight. 12.00	....	97	.06–.065	29	70	600	98	101		2,158	8,139	Midnight 12.00 A. M.	12,479	1,068
A. M. 12.30	....	97	.06–.065	29	72	600	98	100		....		....	....	....
1.00	....	97	.06–.065	30	72	570	98	100		1,544		1.00	12,874	1,098
1.30	....	96	.06–.065	29	72	600	97	100		....	....	....	....	....
2.00	....	97	.06–.065	29	74	675	98	99		2,300	....	2.00	11,892	1,085
2.30	....	96	.06–.065	29	72	660	98	100		....	....	....	....	....
3.00	....	97	.06–.065	29	72	600	98	99		1,968	....	3.00	10,628	1,070
3.30	....	95	.06–.065	29	72	600	97	98		....	....	....	....	....
4.00	....	98	.06–.065	28	72	530	97	98		1,800	....	4.00	12,653	1,077
4.30	....	97	.06–.065	27	68	530	97	98		....	....	....	....	....
5.00	....	98	.06–.065	27	68	600	97	98		1,122	1,834	5.00	12,243	1,071
5.30	....	98	.06–.065	27	68	625	97	98		....		....	....	....
6.00	....	96	.06–.065	26	70	550	97	98		2,000		6.00	11,155	1,076
6.30	....	96	.06–.065	26	68	700	96	97		....	....	....	....	....
7.00	....	96	.06–.065	26	68	700	97	98		2,022	....	7.00	13,826	1,056
7.30	....	97	.06–.065	26	64	700	96	97		....	....	....	....	....
8.00	....	96	.06–.065	26	64	650	96	97		1,580	....	8.00	10,907	1,093
Average 24 hours	30,655 or 15.05 Lbs.	96.53 or 111.58 Absolute.	0.625	35	66	555	96.3	98.2	335,635	44,014 or 1833.92 per hour.	10,088 or 420⅓ per hour.		286,982 or 11,957.5 per hour.	26,275 or 1,094.89 per hour.

The moisture in the coal averaged 6 per cent., which reduces the total number of pounds of coal consumed to 41,373.16 lbs., or 1,723.88 lbs. per hour.

The percentage of refuse was $\frac{10,088}{41,373.16} = 24.38$ per cent.

The total combustible = 31,285.16 lbs. or 1,303.55 lbs. per hour.

The height of water in the gauges was kept almost constant during the test, and was the same at the beginning and ending of the test. Observations were taken every half hour. Twenty observations were taken with the anemometer, giving an average velocity of 235.25 feet per minute, or 235.25′ × 29.3′ = 6,892.825 cubic feet per minute.

RESULTS OF THE TRIALS OF FURNACE-FLUE TUBULAR BOILERS

AT SPRING GARDEN PUMPING STATION, TO DETERMINE DUTY AND CAPACITY OF GASKILL PUMPING ENGINE.

Date of Trial, November 29th and 30th, 1887. Duration of Trial, 24 Hours.

DIMENSIONS AND PROPORTIONS.

Type of boilers: Furnace-flue tubular.	
Number in use	5
Diameter of boiler, feet	8′–6″
Length of boiler, feet	20′
Number of furnaces	2
Type of furnaces, Fox's Corrugated.	
Diameter of furnaces, feet	3′–6″
Length of furnaces, feet	7′–6″
Number of tubes in each boiler	90
Diameter of tubes, inches	4″
Length of tubes, feet	10′
Diameter of drum	....
Length of drum	....
Heating surface in one boiler, sq. ft	1,119
a Heating surface of furnaces (2), sq. ft	136
b Back box	....
c Combustion chamber, sq. ft	47
d Tubes, sq. ft	936
e Front of boiler	....
f Side back	....
Total heating surface in 5 boilers, sq. ft	5,595
Total tube cross section area, sq. ft	8
Ratio of tube cross section to grate surface	1 to 2¼
Height of stack, feet	150
Area of stack at base, sq. ft	25
Grate surface, 3′–6″ wide; 6′ long; 42 sq. ft. area, sq. ft	42
Water heating surface in five boilers, sq. ft	5,595
Superheating surface in five boilers	....
Ratio of water heating surface to grate surface	26⅔ to 1

AVERAGE PRESSURES.

Steam pressure in boiler by gauge, pounds	96.53
Absolute steam pressure, pounds	111.58
Atmospheric pressure per barometer, inches	30.655
Force of draught in inches of water, inches	0.625

AVERAGE TEMPERATURES.

Of external air, degrees	35
Of fire room, "	66
Of steam "	335.635
Of escaping gases "	555
Of feed water, "	96.3

FUEL.

Kind of coal, anthracite pea coal.

Total amount of coal consumed, pounds	44,014
Moisture in coal, per cent	6
Dry coal consumed, pounds	41,373.16
Total refuse (dry waste in pounds), pounds	10,088
Per cent. of refuse	24.38
Total combustible (dry weight of coal, less refuse), pounds	31,285.16
Dry coal consumed per hour, pounds	1,723.88
Combustible consumed per hour, pounds	1,303.55

RESULTS OF ANEMOMETER READINGS.

Cubic feet of air per hour	413,569.5
Cubic feet of air per hour, per square foot of grate	1,969.37
Pounds of air per pound of coal	18.35
Pounds of air per pound of combustible	24.27

RESULTS OF CALORIMETRIC TESTS.

Quality of steam, dry steam being taken as unity	.965
Percentage of moisture in steam	3½
Number of degrees superheated	....
Factor of evaporation	1.16

WATER.

Total weight of water pumped into boiler and apparently evaporated, pounds	286,982
Water actually evaporated, corrected for quality of steam and surface water, pounds	279,578.47
Equivalent water evaporated into dry steam from and at 212° F., pounds	323,888.5
Equivalent total heat derived from fuel in British thermal units	313,154,925.35
Percentage of heat lost by radiation, on a basis of 14,500 B. T. U. per pound of combustible	31
Equivalent water evaporated into dry steam from and at 212° F. per hour, pounds	13,495.35

ECONOMIC EVAPORATION.

Water actually evaporated per pound of dry coal from actual pressure and temperature, pounds	6.75
Equivalent water evaporated per pound of dry coal from and at 212° F., pounds	7.83
Equivalent water evaporated per pound of combustible from and at 212° F., pounds	10.353

COMMERCIAL EVAPORATION.

Equivalent water evaporated per pound of dry coal, with one-sixth refuse, at 70 pounds gauge pressure, from temperature of 100° F. = Item 63 × 0.7249, pounds....................................

RATE OF COMBUSTION.

Dry coal actually burned per square foot of grate surface per hour, pounds...

Consumption of dry coal per hour, coal assumed with one-sixth refuse:

Per square foot of grate surface, pounds........................

Per square foot of water heating surface, pounds............ ..

Per square foot of least area for draught ....

RATE OF EVAPORATION.

Water evaporated from and at 212° F. per square foot of heating surface per hour, pounds ....

Water evaporated per hour from temperature of 100 F. into steam of 70 lbs. gauge pressure = Item 60 × 0.8698 :

Per square foot of grate surface, pounds.........................

Per square foot of water heating surface, pounds................

Per square foot of least area for draught, pounds......... ......

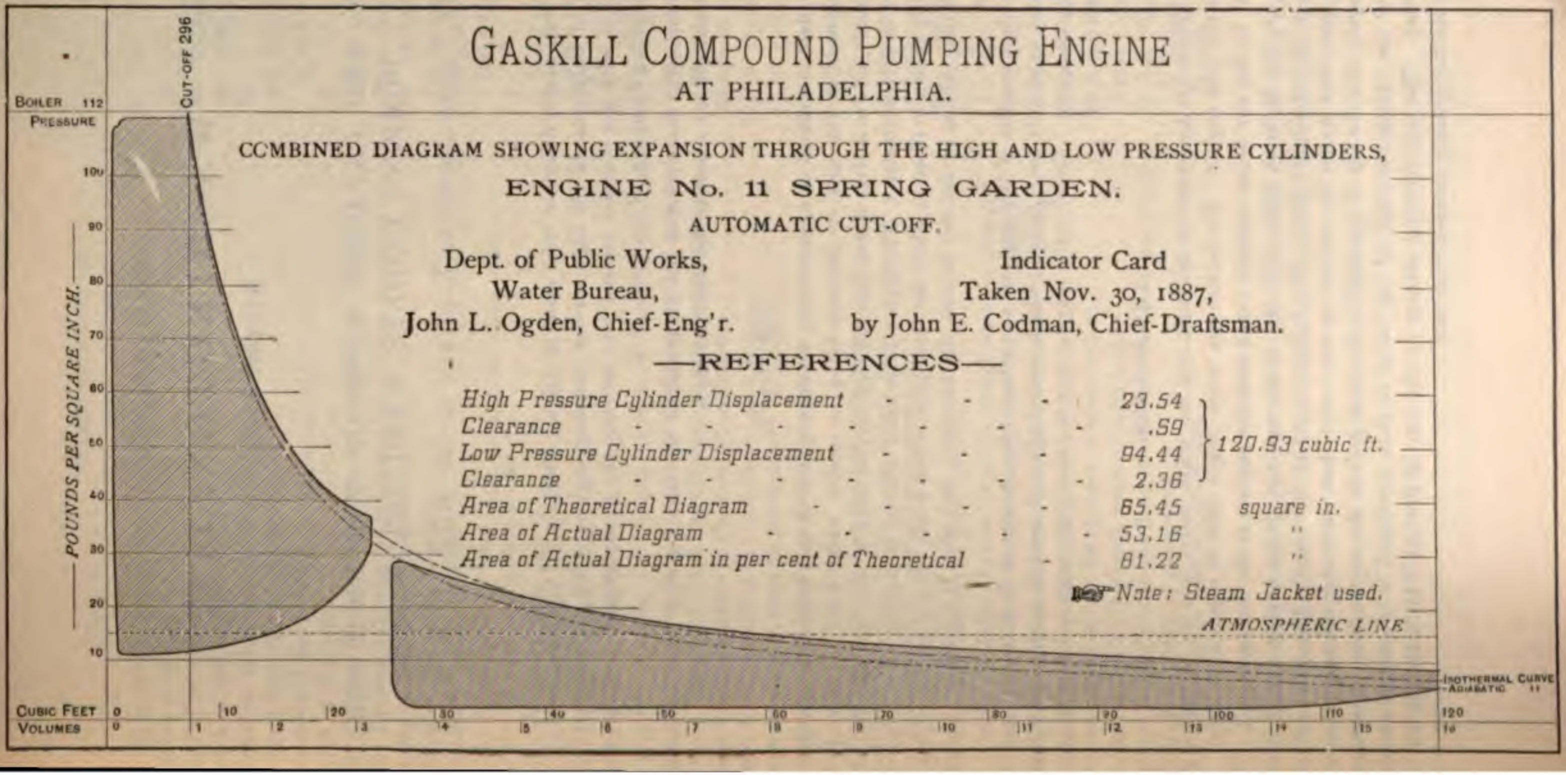
COMMERCIAL HORSE-POWER.

On a basis of thirty pounds of water per hour evaporated from temperature of 100° F. into steam of 70 pounds gauge pressure (= 34½ lbs. from and at 212° F.), H. P........ ....

Horse-power, builders' rating at 10 square feet per horse-power for fire surface, 12 square feet on ⅔ surface of flue and drum, and 15 square feet on ⅔ surface of tubes, H. P........................

Per cent. developed above rating........... ....

Per cent. developed below rating......................... ....



GASKILL COMPOUND PUMPING ENGINE
AT PHILADELPHIA.

COMBINED DIAGRAM SHOWING EXPANSION THROUGH THE HIGH AND LOW PRESSURE CYLINDERS,

ENGINE No. 11 SPRING GARDEN.

AUTOMATIC CUT-OFF.

Dept. of Public Works,
Water Bureau,
John L. Ogden, Chief-Eng'r.

Indicator Card
Taken Nov. 30, 1887,
by John E. Codman, Chief-Draftsman.

—REFERENCES—

High Pressure Cylinder Displacement	23.54	120.93 cubic ft.
Clearance	.59	
Low Pressure Cylinder Displacement	94.44	
Clearance	2.36	
Area of Theoretical Diagram	65.45	square in.
Area of Actual Diagram	53.16	"
Area of Actual Diagram in per cent of Theoretical	81.22	"

☞ *Note: Steam Jacket used.*

WASHINGTON, D. C.

AUGUST, 1888.

DUTY AND CAPACITY TESTS OF THE HIGH SERVICE PUMPING ENGINES, WASHINGTON, D. C.

BY PASSED ASSISTANT ENGINEER G. W. BAIRD, U. S. NAVY.

The tests herein described were made in August, 1888, for the Commissioners of the District of Columbia, on the high service pumping engines at the U Street Station, in accordance with the contract entered into between the Commissioners and the Holly Manufacturing Company, of Lockport, N. Y.

The following description of the plant and of the tests, together with the results, are embodied in my report to the Commissioners:

THE PLANT.

The plant is situated on a reservation between U and S and Sixteenth and Seventeenth streets, in this city, its purpose being to receive water from the 20-inch low service main, and to deliver either into the high service main or into the stand-pipe at the head of Sixteenth street. The high service main referred to supplies all that portion of the city shown in Plate I; while the stand-pipe, which holds 270,919 gallons of water, supplies that portion of the city beyond the boundary, and what is known as Columbia Heights.

The plant consists of two engines, according to the patents of H. F. Gaskill, and two horizontal, externally fired boilers, which were built and erected by the Holly Manufacturing Company. Either boiler may be used to drive either engine. The plants are duplicate.

THE BOILERS.

The boilers are designated as the north and the south boiler, and the engines as the east and the west engine. The boilers are much larger than is essential to drive the engines; in fact one boiler is ample to run both the engines at the contract speed. The boilers are set in brick-work, with the usual cast-iron fronts, having closely fitting furnace doors and ash-pit doors. The furnace is at the front end. The gases of combustion pass first under the bottom of the boiler, then returning through tubes, and return again over the top of the shell to the back end, to the smoke-flue. Almost the whole of the external surface of the boiler is thus enveloped in the heated gases of combustion. The superheating surface on the top of the shell is inefficient, for the reason that it is quickly covered with soot.

rincipal dimensions of the boilers are as follows :

f boilers	2
of the shells	60 inches.
f the shells	14 feet.
f tubes in each boiler	54
diameter of the tubes	3¾ inches.
s of the tubes	⅛ inch.
f the tubes	14 feet.
ance between the tubes and the shell, at the sides	2½ inches.
ance between the tubes and the bottom of the boiler	5½ inches.
f the tubes, vertically	5 inches.
f the tubes, laterally	5¼ inches.
of steam drums	2 feet 9 inches.
the steam drums	3 feet 6 inches.
f furnaces to each boiler	1
the furnace grates	5 feet 6 inches.
the furnace grates	5 feet 6 inches.
rate surface in each boiler	30.25 square feet.
rate surface in both boilers	62 5 square feet.
e cross area, for draught through the tubes in each boiler,	3.71 square feet.
e cross area, for draught over the bridge-wall of each boiler,	4.03 square feet.
raught through the chimney, at smallest contraction	3.82 square feet.
ating surface in the shell of each boiler	63 square feet.
ating surface in the tubes of each boiler	742 16 square feet.
ating surface on the heads of each boiler	14.93 square feet.
e water heating surface in each boiler	820.09 square feet.
om in each boiler	139.28 cubic feet.
ting surface in the shell of each boiler	54.978 square feet.
ting surface in the drum of each boiler	15.119 square feet.
e superheating surface in each boiler	70.097 square feet.
m in each boiler	69.6 cubic feet.
water heating to grate surface	27.11 to 1.
rate surface to area over the bridge-wall for draught	7.5 to 1.
grate surface to area through the tubes for draught	8.15 to 1.
grate surface to area through the chimney for draught	7.92 to 1.
chimney above the grate	88 feet 4 inches.
length of the brick work of each boiler	20 feet 6 inches.
width of the brick work of each boiler	8 feet 7 inches.
height, above the floor, of the brick work of one boiler	11 feet 7 inches.
top edge of grate bars	1⅝ inches.
air space between grate bars	⅝ inch.

COAL.

el consumed during the test was a good quality of George's Creek Cumber-
i-bituminous) coal, of the kind regularly shipped by the contractor, and was
st as received.

ial analysis by A. S. McCreath gave the following :

ric water	1.01
atter, other than water	13.92
bon	82.96
	3.11

THE ENGINES

There are two separate engines shown in plate [see frontispiece and following plates], each having four steam cylinders and two water cylinders. The engines are horizontal, compound condensing engines and are in pairs; each pair acting on a crank placed at right angles on a single shaft. There is a fly-wheel on the middle of the shaft.

The high-pressure cylinders are placed on top of the low-pressure cylinders, with very short steam passages between them.

The piston-rod of the high-pressure cylinder is connected, through links, to the upper end of a working-beam, and the two piston-rods of the low-pressure cylinders are connected similarly to the lower end of the same beam. The movements of the pistons of each pair are therefore co-existing and alternate. From the cross-heads of the high-pressure cylinders the connecting-rods extend to and work the cranks, and from the cross-heads of the low-pressure cylinders the pumps are worked.

The rock-shafts, the beam-centers, extending from the beams, carry arms which work the air and feed-pumps.

The valves are actuated by eccentrics on revolving-shafts, which receive their motion from the main shaft through the intervention of miter-gearing. The opening of the high-pressure steam valves which are poppets is effected by the intervention of [illegible] [illegible] with an intervening lever. By an unique arrangement the one toe knocks off the other effecting a trip cut-off. A dash-pot is employed to cushion the valve in its fall.

The intermediate valves and the exhaust valves of the low-pressure are gridiron valves. The valves are readily accessible for repairs. The cut-off (the h. p. steam) valves are momentarily adjustable.

The steam jackets, which cover the heads and sides of all the steam-cylinders, drain into a coil of feed-water heater, the supply of steam being regulated by valves, without the interposition of steam traps.

The cross-head guides are single, rectangular bars and the cross-heads are of the locomotive pattern.

The links and connecting-rods are fitted with the ordinary gibs and keys, and the pillow blocks with the ordinary caps and bolts.

The beam centers are carried on cast iron pedestals bolted to the sole plates, and are braced, from the steam cylinder on the one side and the water cylinder on the other, by wrought iron braces having collars and keys. The engines are mounted on brick foundations.

The water end is of the usual form, except that it has a plunger (after the Worthington style) which is packed by means of a stuffing box and gland; this packing is reached only by removing the cylinder head of the pump. The water valves are of bronze, with rubber seats. The net opening of each valve is 1.35 square inches and its weight is 15 ounces.

There is an air chamber to each water end. A No. 8 Westinghouse air pump is provided to supply air to these chambers when necessary.

There is a surface condenser for each engine. The condensers are cylindrical in form, with their axes horizontal, they each contain 128 brass tubes one inch outside diameter, and seven feet five inches in length. The circulating water passes through the tubes and the circulation is derived by a shunt from the main suction.

There are four single-acting air pumps on each engine, having a diameter of ten inches and a stroke of twelve inches. The aggregate piston displacement of the air pumps, per revolution, is 2.18 cubic feet. The air pump valves are eight inches diameter. There are four single-acting piston feed pumps on each engine, worked from arms on the beam center; they receive their water from the feed water heater above described, which receives its heat from the drains from the jackets.

The main shaft is carried on pillow blocks, which are cast on the discharge chambers of the main pumps.

The fly-wheel, which is on the middle of the main shaft, runs between the two pump cylinders.

The pumps receive their water from a twenty-inch main, and discharge into the high service mains against a head of about fifty-seven feet ordinarily.

The following are the principal dimensions of one engine :

Number of high-pressure cylinders	2.
Diameter of the high-pressure cylinders	12 inches.
Diameter of the high-pressure piston rods	1⅞ inches.
Net area of each high-pressure piston	111.71
Stroke of the high-pressure pistons	22 inches.
Displacement of the high-pressure piston per stroke	1.4222 cubic feet.
Space in clearance and steam passages at one end of high-pressure cylinder	.035 cubic feet.
Ratio of clearance to piston displacement	2.461 per cent.
Area of steam port of high-pressure cylinder	7⅛ square inches.
Area of exhaust port of high-pressure cylinder	12¼ square inches.
Clearance of high-pressure piston	⅛ inch.
Number of low-pressure cylinders	2.
Diameter of the low-pressure cylinders	24 inches.
Diameters of the (two) low-pressure piston rods	2¼ inches.
Net area of each low-pressure piston	448.41 square inches.
Stroke of the low-pressure pistons	22 inches.
Displacement of each low-pressure piston for one stroke,	5.709 cubic feet.
Space in clearance and steam passages at one end of low-pressure cylinder	.17 cubic feet.
Ratio of clearance to piston displacement	2.96 per cent.
Area of low-pressure steam port	12¼ square inches.
Area of low-pressure exhaust port	24½ square inches.
Clearance of low-pressure piston.	
Number of air pumps	4.
Diameter of each air pump	10 inches.
Net area of the valves of each air pump	47 square inches.
Stroke of the air pumps	12 inches.
Aggregate piston displacement of the air pumps of one engine for one revolution	2.18 cubic feet.
Number of feed pumps	4.
Diameter of the feed pumps	2¼ inches.
Stroke of the feed pumps	6 inches.
Aggregate displacement of the feed pump pistons for one revolution of the engine	0.055 cubic feet.
Diameter of the condenser	24½ inches.
Length of the condenser	8 feet 4 inches.
Number of tubes in the condenser	128.
External diameter of the condenser tubes	1 inch.
Length of the condenser tubes	7 feet 5 inches.
Aggregate condensing surface	248.5 square feet.

Square feet of condensing surface per indicated horse-power	3.17
Length of the connecting rods between centers	64 inches.
Area of the neck of each connecting rod	5 square inches.
Diameter of the cross-head journals, H. P.	2¼ inches, L. P., 2½ inche
Length of the cross-head journals, H. P.	2½ inches, L. P., 3 inches.
Diameter of the crank pin journals	3 inches.
Length of the crank pin journals	3¾ inches.
Number of main journals on each shaft	2.
Diameter of the main journals	6 inches.
Length of the main journals	11 inches.
Diameter of the beam-center journals	5 inches.
Length of the beam-center journals	9 inches.
Diameter of the fly-wheel	7 feet 11 inches.
Area of cross section of the rim of the fly-wheel	48 square inches.
Weight of the fly-wheel	7200 lbs.
Diameter of the pump plungers	14½ inches.
Diameter of the pumps rods	2½ inches.
Net area of each pump plunger	162.68 square inches.
Net displacement of each plunger, per stroke, in gallons,	15.493
Aggregate displacement of the two pump plungers for one revolution of the engine	14318.62 cubic inches.
Number of suction valves in each end of each pump	35.
Diameter of opening of each valve	$1\frac{5}{16}$ inches.
Aggregate area of opening of the suction valves for each end of each pump	47.25 square inches.
Ratio of area of plunger to area through the suction valves	3.44 to 1.
Number of discharge valves in each end of each pump	34.
Diameter of opening of each valve	$1\frac{5}{16}$ inches.
Aggregate area of opening of the discharge valves of each end of each pump	45.90 square inches.
Ratio of area of plunger to area through the discharge valves	3.54 to 1.
Number of air chambers on each engine	2.
Diameter of the air chambers	2 feet.
Height of the air chambers	4 feet 6 inches.

THE TERMS OF THE CONTRACT.

According to the terms of the contract the duty test was to continue for twenty fo hours consecutively, during which time it was to work smoothly and steadily, witho heating or vibration, at the contract rate of delivery, which is 2,500,000 gallons water against a head of 140 feet, at a piston speed not exceeding 110 feet per minu The contract requires a duty of 90,000,000 foot-pounds for 100 pounds of coal "bas on an evaporation of ten to one." The contract also provides that the computation f capacity shall be made from the plunger displacement, making due allowance for sl as the Engineer Commissioner shall determine.

The contract further provides that the duty shall be computed by the formula

$$\text{Duty} = \frac{P\ N\ H \times 100}{W} \qquad [1.]$$

in which P = pounds of water delivered per stroke.

N = total number of strokes.

H = total fluid resistance, including static and frictional heads, and allowing one pound for friction through the pumps and passages.

W = number of pounds of coal burnt during the test, not computing the coal used for raising steam.

By the term "an evaporation of ten to one" I understand, in this case, a vaporization of ten pounds of water for each and every pound of coal, from the temperature of the feed water, and at the temperature due to the boiler pressure employed.

The amount of water vaporized by a given weight of coal is universally conceded to be the proper measure of the efficiency of a boiler, but in the comparison of boilers, they should have equally good coal, be fed with water of the same temperature and furnish steam at the same pressure. The comparison is properly made by reference to the *equivalent vaporization from and at* 212 *degrees per pound of combustible*; that is, what the vaporization would have been had the coal been without ashes or refuse of any kind; the feed water delivered to the boiler at the boiling point and vaporized under the atmospheric pressure alone.

By reference to the contracts for pumping engines, between E. P. Allis & Co. with the city of Hannibal, Mo., with the city of Allegheny, Pa., and with the city of Decatur, Ill., and Robert Wetherill & Co. with the city of Philadelphia, Pa., and with Cincinnati, Ohio, I find the stipulation of a vaporization of ten pounds of water per pound of coal, as in this case, and that the contractors understand it to mean, unequivocally, that their duty is to be based on a vaporization of ten pounds of water for each and every pound of coal, from the normal temperature of the feed water.

MANNER OF MAKING THE TESTS.

The scales for weighing the feed-water and for weighing the coal were tested by the sealer of weights and measures, and found to be correct.

The pressure gauges were compared with a certified gauge, and corrections made for them where errors were found. The thermometers agreed with the standard. The indicators, which were borrowed from the Bureau of Steam Engineering of the Navy, were tested and verified.

The indicators used in testing the west engine were of the Thompson make, and those used in testing the east engine—which came later—were of the Tabor make.

The coal was weighed by clerks detailed from the commissioner's offices, as it came from the bunker. It was George's Creek Cumberland coal, the kind furnished by the contractor, and was used just as it came; there was no attempt at selecting the better portion.

A pyrometer was placed in the up-take; its error, by comparison with thermometers at 70 and 210 degrees, showed a constant error—in excess—of 241 degrees. The record of this temperature is, therefore, that shown by this instrument, minus 241.

The pressure in the mains was indicated by pressure gauges, which were level with the pump barrels, center to center, and about three feet from the pumps. The pressure in the rising mains was measured by a similar gauge, the distance of which, below the center line of the pump, was accurately ascertained by leveling.

The feed-water was weighed in a cask, which emptied into a second cask, from which latter it was drawn by the feed-pumps.

The contractors' experts ran the engines and the boilers to suit themselves, without interference on either my part or that of the district employees. The coal was weighed and delivered to the fire-room in charges of 250 pounds each, and the firemen were instructed to regulate the fire and the draughts so as to keep a pressure on the boiler gauge of nearly 100 pounds. The ash-pit doors were kept closed nearly all the time, to reduce the rate of combustion. The ashes and the clinkers were weighed dry. As the engine and the boiler continued on their regular duty, the fires were not hauled at the end of the tests. From the weights of ashes thus obtained (about 5 per cent.) it is evident that, in cleaning the fires, the fair proportion had not been withdrawn. The average percentage of ashes for the ten days preceding and succeeding the tests was exactly ten per centum of the coal used. This quantity will, therefore, be used in the calculation for evaporation.

The steam-jackets were not trapped, and there was no means of knowing how much live steam was blown through them into the feed-water; they discharge into the feed-water heater (and into the feed-water), the supply being regulated by throttling the steam *before* it reached the jackets. The extraordinary temperature of the feed-water would indicate that live steam was freely blown into it.

The head of water against which the engines worked when pumping into the high service mains, was about 57 feet, but when delivering into the stand-pipe, about 140 feet. Usually the pumps are not required to pump into the stand-pipe for a longer period than two hours daily. The resistance of 140 feet, stipulated in the contract, was obtained by partially closing a stop valve in the main, about 50 feet from the pump-house. This valve was regulated by the superintendent of the water works. During the test of the west engine an indicator diagram was taken each hour from each end of each steam cylinder and each water cylinder. During the test of the east engine the diagrams were taken half-hourly. During both tests the pressure in the boilers, the vacuum in the condensers and the pressure in the mains were observed every 15 minutes, averaged and recorded at the end of the hour. The counter, the temperatures of the main, the rising main, the feed-water, the up-take and the atmosphere, and the height of the barometer were noted hourly; the quantity of coal and the casks of feed-water were charged as they were weighed.

The same boiler was used for both tests; it had been in operation about five days before the first test was begun, and its tubes had not been swept during that period. After completing the first, and before commencing the second test, the tubes were swept.

At the moment of commencing the tests the counter was noted, the condition of the fire was observed, and the height of the water in the gauge glass on the boiler was marked in the usual way. The tests continued, each precisely 24 hours, uninterruptedly, during which time the steam pressure did not vary more than three pounds nor the vacuum more than an inch of mercury; the throttles remained wide open, th engines being regulated by the cut-off.

The speed varied between 30.1 and 32.5 revolutions per minute. An undue hammering in the water end of the west engine at the end of the thirteenth hour wa caused, no doubt, by the pressure on the suction side, which had risen to seven pounds it ceased when that pressure fell. It could have been stopped by partially closing th valve in the rising main. To prevent this hammering, under such conditions, it is th practice of some of the best builders to put springs on the suction valves.

The feed-pump of the west engine, soon after commencing the test, stopped working; it was probably full of steam (from the jackets). The contractors' expert relieved the pressure on the pump by permitting a small quantity of the feed-water t escape; the pumps gave no further trouble, except requiring frequent tightening o the plunger packings.

The Westinghouse air-pump was used two minutes, only, during the 24-hour test of the west engine, and not at all during the test of the east engine.

During the test of the west engine the safety-valve lifted, and a quantity of steam escaped; it was closed as quickly as possible. No account could be made, nor was credit given for the loss of water or heat by this accident, nor did the contractors' experts claim any.

Passed Assistant Engineers Richard Inch and C. W. Rae, whom the Navy Department detailed for the purpose, stood regular watches with the writer, taking the indicator diagrams, making the observations and recording the data.

The slip was measured by stopping one engine and putting the normal pressure of the high service mains upon the discharge valves and on one side of the plunger of one of the pumps (which was taken at random); the space on the other side of the plunger was piped to a cask, in which the leakage was weighed. The valves from the rising main to the pump were tightly closed and, being new, were assumed to be tight. In this way all the water which passed through the plunger or through the discharge-valves was trapped and weighed. The capacity of the cask was 992 pounds, so the duration of the test was short. A mean of five tests gave a slip of 0.96 per cent. of the plunger displacement, at a mean head of 52.49 feet, and calculated at the plunger displacement at the time of the duty test. The slip will vary as $\sqrt{h}$, so that the mean slip for the west engine will be

$$\frac{\sqrt{143.02} \times 0.96}{\sqrt{52.49}} = 1.58 \text{ per cent., and for the east engine}$$

$$\frac{\sqrt{144.03} \times 0.96}{\sqrt{52.49}} = 1.59 \text{ per cent.}$$

To check this measurement of the slip, I had the valves leading to the distributing mains of the high service closed, and the water pumped directly into the stand-pipe, a vessel holding 270,919 U. S. gallons, located on a hill about 2,000 feet from the pumps. Two observers were placed on the stand-pipe, and, by an electric signal, announced to the observers in the engine-house the instant the water began to flow on the floor of the stand-pipe, and again when it reached a given height. The observers in the engine-house noted the counter at each signal and noted, at intervals of 15 minutes, the pressure indicated by the gauge on the mains. From the data thus obtained the slip appeared to be six per cent. The engine was then stopped, when it was observed that the head diminished from 159.4 to 90 feet in about fifteen minutes, due, no doubt, to a leak either in the main or in the valve. There is a check-valve which holds the water in the stand-pipe; consequently the diminution of pressure was in the pipe alone, which accounts for the rapid diminution. The existence of this leak of course barred the test for slip. The original measurement was, therefore, accepted and assumed to be correct.

The engines were taken off their regular duty, several days after the duty test, and the pump heads taken off, the suction and discharge valves having been closed. The engines were then started and three complete sets of indicator diagrams taken, to ascertain the power required to work each engine and its concomitants. A jet was used in the condensers to condense the steam; all the air pumps and feed pumps (four of each to each engine,) remained connected. The engines were run at the same speed, as nearly as possible, as on the days of the duty tests. The extraordinary pressure per square inch of piston required to work the engines (about 9 pounds in the high pressure and about 0.8 in the low), is easily understood when it is remembered that each engine has four air pumps, four feed pumps, six piston rods, two pump rods, twelve valve rods, four steam pistons and two water plungers, besides the beam centers, crank-shaft bearings, cross-head guides, etc., to interpose friction.

Capacity of the west engine :

$$15.493 \times 4 \times 44585\,(1-0.0158) = \mathbf{2{,}719{,}366}\ \textbf{gallons}\ \text{of water in 24 hours.}$$

Duty of the west engine :

$$\frac{126.64 \times 4 \times 44585 \times (143.02-7+3.55) \times 100}{4123} = 76{,}453{,}682$$

foot pounds of water lifted for 100 pounds of coal.

Duty of the west engine referred to the contract basis of 10 pounds of water vaporized per pound of coal, and the addition of one pound (2.31 feet) to the head :

$$\frac{126.64 \times 4 \times 44585 \times (143.02-7+3.55+2.31) \times 10 \times 100}{4123 \times 7.681} = \mathbf{101{,}183{,}587}$$

foot pounds per 100 pounds of ccal.

Rate of delivery at contract speed of piston of 110 feet per minute ;

$$15.493 \times (1-0.0158) \times 4 \times 30 \times 1440 = 2{,}634{,}890\ \text{gallons in 24 hours.}$$

Capacity of the east engine :

$$15.493 \times 4(1-0.0159) \times 44354 = \mathbf{2{,}750{,}002}\ \textbf{gallons}\ \text{of water in 24 hours.}$$

Duty of the east engine :

$$\frac{126.71 \times 4 \times 44354\,(144.03-5+3.55)}{3862} = 82{,}998{,}124$$

foot pounds per 100 pounds of coal.

Duty of the east engine referred to the contract basis of 10 pounds of water vaporized per pound of coal, and the addition of one pound (2.31 feet) to the head :

$$\frac{126.71 \times 4 \times 44354\,(144.03-5+3.55+2.31) \times 10 \times 100}{3862 \times 8.287} = \mathbf{101{,}772{,}977}$$

foot pounds per 100 pounds of coal.

FULFILLMENT OF THE CONTRACT.

The engines worked during the whole of the period of the tests very steadily, without any heating or jarring, though running above the contract speed. I have no doubt they can be run, continuously, 20 per cent. above that speed without injury.

The *duty* of the west engine exceeded the requirements by 14.25 per cent. ; the duty of the east engine exceeded the requirements by 14.85 per cent.

The capacity of the engines at 110 feet piston speed per minute exceeded the requirements by 5.39 per cent. in the west engine and by 5.38 per cent. in the east engine.

REMARKS.

The large amount of heating surface in the boiler, in proportion to the area of the grate, and the low rate of combustion would premise a high economic efficiency of the boiler. The comparative failure of the boiler, in this respect, is due, I have no doubt, to the in-leak of cold air through the brick work which envelopes the boiler. Had the boiler been more efficient the duty of the engine, *ceteris paribus*, would have been greater.

The method adopted by the contractor of measuring the steam jackets is not the best; it would be better to provide proper traps and to keep the boiler pressure in the jackets when they are used. This will have the effect of utilizing the heat in the steam cylinders—where it is most needed—and, when properly trapped, will deliver water and not steam to the feed pumps.

The engines would be improved by substituting hard brass for the cast iron feed pump plungers, which seem to cut out the packing rapidly; the stuffing boxes on these plungers would be improved if made deeper.

The gearing for adjusting the point of cutting off, which is used to regulate the ed of the engine, is inconvenient and is not easy to manage; a spiral gear could be stituted for the present hand lever at a moderate expense.

The quality of the material, the character of the castings and forgings, and other ing and finish of the engine are fully equal to that in first-class pumping engines. able of the data and results is appended:

BLE CONTAINING THE DATA AND RESULTS OF THE DUTY AND CAPACITY TESTS OF THE HIGH SERVICE PUMPING ENGINES OF WASHINGTON CITY IN THE MONTH OF AUGUST, 1888, BY G. W. BAIRD, PASSED ASSISTANT ENGINEER, U. S. NAVY.

	WEST ENGINE.	EAST ENGINE.
Date of commencing the test	8 Aug., 1888.	13 Aug., 1888.
TOTAL QUANTITIES.		
Duration of the tests, in hours	24.	24.
Total number of revolutions made by the engines	44,585.	44,354.
Total number of pounds of feed water pumped into the boiler	31,671.	32,007.
Total number of pounds of coal consumed	4,123.	3,862.
Total number of pounds of ashes	224.	163.
Total number of pounds of combustible consumed	3,899.	3,699.
Total number of gallons of water lifted	2,719,366.	2,705,002.
MEAN QUANTITIES.		
Steam pressure in the boiler in pounds per square inch above the atmosphere	95.81	95.81
Fraction completed of the stroke of the H. P. piston, when the steam was cut off	0.2618	0.2768
Number of times the steam was expanded	18.01	16.95
Height of the barometer in inches of mercury	29.978	30.09
Vacuum in the condenser in inches of mercury	25.260	24.80
Pressure in the condenser in pounds per square inch, above zero	2.316	2.597
Head of water, in feet, in the main	143.02	144.03
Head of water, in feet, in the rising main	7.	5.
Number of revolutions, per minute, of the engine	30.961	30.801
Number of pounds of feed water pumped into the boiler per hour	1,319.625	1,333.625
Number of thermal units imparted to the feed water per hour	1,294,565.3	1,319,141.6
Height of center line of pump above center of pressure gauge on rising main, in feet	3.55	3.55
MEAN TEMPERATURES.		
Temperature of the atmosphere	89.12	77.20
Temperature of the water in the rising main	83.30	80.58
Temperature of the water in the main	85.50	81.50
Temperature due to the steam	334.92	334.92
Temperature of the feed water	203.58	196.20
Temperature of the up-take	411.3	326.6

		West Engine.	Eas
27.	Total number of pounds of water lifted	22,584,662.	22,480,
28.	Mean number of pounds of water lifted per hour.	941,043.	941,
29.	Mean number of pounds of water lifted per stroke	126.64	
	STEAM PRESSURES PER INDICATOR.		
30.	Steam pressure on the high pressure pistons at the commencement of the stroke, in pounds per square inch above zero	108.986	
31.	Steam pressure on the high pressure pistons, in pounds per square inch above zero, at the end of the stroke	36.945	
32.	Mean back pressure against the high pressure pistons during its stroke, in pounds per square inch above zero	15.956	
33.	Mean pressure per square inch on the high pressure pistons required to work the unloaded engine	8.9583	
34.	Indicated pressure on the high pressure pistons, in pounds per square inch	53.826	
35.	Net pressure on the high pressure piston, in pounds per square inch	44.868	
36.	Total pressure on the high pressure pistons, in pounds per square inch above zero	69.782	
37.	Pressure on the low pressure pistons at the commencement of the stroke, in pounds per square inch above zero	27.85	
38.	Pressure on the low pressure pistons at the ends of the strokes, in pounds per square inch above zero	10.667	
39.	Mean back pressure against the low pressure pistons during the stroke, in pounds per square inch above zero	4.166	
40.	Mean pressure per square inch on the low pressure pistons required to work the unloaded engines	0.7375	
41.	Indicated pressure on the low pressure pistons, in pounds per square inch	11.135	
42.	Net pressure on the low pressure pistons, in pounds per square inch	10.398	
43.	Total pressure on the low pressure pistons, in pounds per square inch above zero	15.301	
	HORSE-POWER.		
44.	Indicated horse-power developed in the two high pressure cylinders	41.368	
45.	Indicated horse-power developed in the two low pressure cylinders	34.351	
46.	Aggregate indicated horse-power developed by the engine	75.719	
47.	Net horse-power developed in the two high pressure cylinders	34.485	

	WEST ENGINE.	EAST ENGINE.
48. Net horse-power developed in the two low pressure cylinders	32.043	34.916
49. Aggregate net horse-power developed by the engine	66.528	69.089
50. Total horse-power developed in the high pressure cylinders	53.639	54.934
51. Total horse-power developed in the two low pressure cylinders	47.223	48.205
52. Aggregate total horse-power developed by the engine	100.862	103.139
CONSUMPTION OF FEED WATER.		
53. Number of pounds of feed water consumed per hour per indicated horse-power developed by the engine	17.569	16.878
54. Number of pounds of feed water consumed per hour per net horse-power developed by the engine	19.835	19.303
55. Number of pounds of feed water consumed per hour per total horse-power developed by the engine	13.083	12.930
56. Thermal units consumed per hour per indicated horse-power developed by the engine	17,096.967	16,695.247
57. Thermal units consumed per hour per net horse-power developed by the engine	19,459.83	19,093.36
58. Thermal units consumed per hour per total horse-power developed by the engine	12,835.015	12,789.93
STEAM ACCOUNTED FOR BY THE INDICATOR.		
59. Pounds of steam present per hour in the high pressure cylinders at the end of the stroke of the pistons, calculated from the pressures there	1,089.4507	1,186.8906
60. Pounds of steam pressure per hour in the two low pressure cylinders, at the end of the strokes of the pistons, calculated from the pressures there	1,221.5554	1,235.1864
61. Difference, in pounds, per hour between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicators, in the two high pressure cylinders at the ends of the strokes of the pistons	230.174	146.734
62. Difference, in per cent., of the weight of the water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the high pressure cylinders at the end of the strokes of the pistons	17.442	11.002
63. Difference, in pounds per hour, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicators, in the two low pressure cylinders at the ends of the strokes of the pistons	98.070	98.439

		WEST ENGINE.	EAST ENGINE.
64.	Difference, in per cent., of the weight of the water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the low pressure cylinders at the ends of the strokes of the pistons	7.431	7.381
65.	Per cent. which the aggregate indicated horse-power is of the aggregate total horse-power developed by the engine	75.071	76.608
66.	Per cent. which the aggregate net horse-power is of the aggregate total horse-power developed by the engine	65.959	66.986
	VAPORIZATION.		
67.	Mean number of pounds of water vaporized per hour	1,319.625	1,333.625
68.	Total number of pounds of water that would have been vaporized in the boiler had the feed water been supplied at a temperature of 100 degrees Fahrenheit and vaporized under the atmospheric pressure	29,137.32	30,246.61
69.	Total number of pounds of water that would have been vaporized in the boiler had the feed water been supplied at 212 degrees and vaporized under the atmospheric pressure	33,222.879	33,767.385
70.	Number of pounds of water vaporized from 100 degrees by one pound of coal	7.067	7.831
71.	Number of pounds of water vaporized from 212 degrees by one pound of coal	8.057	8.743
72.	Number of pounds of water vaporized from 100 degrees by one pound of combustible	7.852	8.702
73.	Number of pounds of water vaporized from 212 degrees by one pound of combustible	8.953	9.675
74.	Number of pounds of water vaporized from the normal temperature of the feed for one pound of coal	7.681	8.287
	RATE OF COMBUSTION.		
75.	Number of pounds of coal consumed per hour	171.791	160.916
76.	Number of pounds of combustible consumed per hour	154.875	144.812
77.	Number of pounds of coal consumed per square foot of grate per hour	5.679	5.319
78.	Number of pounds of combustible consumed per square foot of grate per hour	5.119	4.708
79.	Number of pounds of coal consumed per square foot of heating surface per hour	0.209	0.196
80.	Number of pounds of combustible consumed per square foot of heating surface per hour	0.188	0.176

BOSTON, MASS.

AUGUST AND SEPTEMBER, 1888.

WILLIAM JACKSON, CITY ENGINEER.

[The following report of tests of engines is from Document 38—1889, **City of Boston**, Twenty-second Annual Report of the City Engineer for the year 1888, pages 22 to 36]:

TESTS OF GASKILL PUMPING ENGINES AT CHESTNUT HILL.

On August 9th, 10th, August 22d and 23d, and September 26th, 27th, tests were made to ascertain the duty of the new high-service pumping engines at Chestnut Hill, and also to determine the efficiency of the boilers connected with them.

The engines are of the type known as the Gaskill Horizontal Pumping Engine, and were built by the Holly Manufacturing Company of Lockport, N. Y.

The general specifications under which bids were received were prepared by the City Engineer, in June, 1886, and the following extracts from them will show the capacity and duty required of the engines, and the methods prescribed for ascertaining the same:

The engines are each to have sufficient capacity to easily deliver into the Fisher Hill Reservoir, so called, eight million (**8,000,000**) **U. S. gallons** of water per twenty-four hours, the total lift being from 116 to 128 feet.

The loss of action in the pumps must not exceed four per cent.

The piston speeds per minute, for the above-named capacities, are not to exceed, for direct-acting engines, one hundred and ten (110) feet, and for beam and fly-wheel engines (vertical), two hundred (200) feet.

As the efficiency of the engines is required to be shown independently of that of the boilers, the calculation of the required duties of the engines will be based upon a unit of water evaporated into dry steam in the boilers, the equivalent of which shall be eleven hundred (1,100) pounds of water evaporated from and at 212 degrees Fahrenheit.

Each engine will be required to exhibit a duty of **one hundred million (100,000,000) foot-pounds** per before specified unit of water evaporated into dry steam in the boilers.

The duty trials, one for each engine, will be conducted in the following manner: steam will be raised in the boilers to the lowest pressure, at which the engines will run at full speed for capacity, when the fires will be hauled, the furnaces and ash-pits cleaned, and fresh fires started with as little delay as possible. As soon as the fires are well ignited the engine will be started and run at full speed for capacity, with the working pressure of steam, for twenty-four hours. At the latter part of the trial the fires will be burned down until the steam-pressure in the boilers is the same as at the beginning of the trial, when the engine will be stopped. The amount of water in the boilers is to be the same at the end as at the commencement of the trial. During the trial the pressure and temperature of the steam, and the temperature and weight of water supplied to the boilers, will be carefully noted. Calorimeter tests will be made to determine the quality of the steam supplied to the engines, and steam containing not more than three per cent. of entrained water will be considered dry steam.

The duty of the engine will be computed as follows: The area of the plunger in feet will be multiplied by the pressure in pounds, due to the total lift in feet, which will be obtained from the average reading of a correct pressure-gauge, attached to the force-main, near its connection to the pump, to which will be added the pressure due to the difference of level between the zero of the gauge and the water in the pump-well; the product thus obtained will be futher multiplied by the total number of feet traveled by the plungers on their discharging strokes. This product equals the total work in foot-pounds. The amount of water evaporated by the boilers during the trial will be reduced to an equivalent number of pounds of water evaporated from and at 212 degrees Fahrenheit. The total work in foot-pounds divided by this equivalent evaporation, and the quotient multiplied by 1,100 will be duty of the engine in foot-pounds, per specified unit of water evaporated into steam. All steam used for feed-pumps, condensers, and other appliances will be taken from the boiler furnishing steam to the engine, and no allowances will be made for steam so used.

The efficiency of the boilers will be tested at the duty trials.

Capacity tests of the engines will be made either in conjunction with the duty trials or separately; the amount of water pumped to be determined by weir-measurement at the Fisher Hill Reservoir.

The duty trials will be conducted by the City Engineer, and the expense of them will be borne by the city.

The contractor, or his representatives, may be present at the duty and capacity trials of the engines, and may have access to all instruments used and notes taken.

The water fed to the boilers included water from the force main and the condensed water from the cylinder-jackets, and was weighed in a barrel placed upon a tested platform scale. After weighing, the water was run into a large tub, from which the feed-pumps were supplied. In the test of engine No. 1, a Knowles feed-pump was used for pumping the feed-water, and in the test of engine No. 2, a feed-pump on the engine was used. The measurement of the feed-water was checked by a Worthington meter placed in the boiler feed-pipe.

The approximate quantity of water returned from the cylinder-jackets was obtained by metering the supply from the force main, and subtracting this amount from total amount of feed-water used.

The tests were made under the direction of the City Engineer, and care was taken to secure accuracy in all observations. The observers were divided into night and day parties, and the more important readings were taken independently by two persons. The tests were under the immediate supervision of Mr. JOHN E. CHENEY, Assistant City Engineer, and Mr. DEXTER BRACKETT, Superintendent Eastern Division Boston Water Works, the former taking charge of the night party, and the latter the day party. Assistant Engineer John A. Gould acted as principal assistant to the above mentioned.

Calorimeter tests of the steam were made during the first and second tests by Mr. George H. Barrus, M. E. On account of the dryness of the steam shown by these tests, and the large amount of steam used in making them, it was not considered advisable to continue these tests at the third trial.

The weir measurements at Fisher Hill Reservoir were made under the direction of Mr. Desmond FitzGerald, Resident Engineer and Superintendent Western Division Boston Water Works.

It will be noted that, in accordance with the specifications, the duty of the engines was to be computed upon a unit of steam used by them, instead of upon the usual unit of coal burnt under the boilers. The duty of the engines was therefore to be shown in terms of heat furnished to and consumed by them. The unit ot steam used was made large, and consequently no allowances were made for any steam not used by the engine proper, nor for any heat returned from the engine to the boiler.

The duty required by the specifications can be expressed in general terms as:

Duty per 1,100 pounds water evaporated from and at 212°

$$= \frac{\text{Work done in foot-pounds} \times 1{,}100}{\text{Equivalent pounds of water evaporated from and at } 212^\circ.}$$

At the trial of engine No. 2, made on September 26th, 27th, the work done was 8,754,116,000 foot-pounds, and the weight of water fed to the boiler and evaporated into steam at 79.9 pounds pressure, was 88,453 pounds. The equivalent evaporation from and at

$$212^\circ \text{ was } 88{,}453 \times \frac{1{,}000}{966} = 91{,}566.25 \text{ pounds.}$$

Then,—

Duty per 1,100 pounds water evaporated from and at 212°

$$= \frac{8{,}754{,}116{,}000 \times 1{,}100}{91{,}566.25} = \mathbf{105{,}164{,}600 \text{ foot-pounds}}$$

The specified unit of water evaporated into steam equals 1,100 pounds of steam made from and at 212°, and this can be expresed by the 1,062,600 heat units contained therein By adapting the formula to this expression of the specified unit, it becomes general in character and application, and appears in the following form:

Duty per 1,100 pounds steam made from and at 212°

$$= \frac{W \times 1{,}062{,}600}{U - u} \text{ in which}$$

W = Work done by engine in foot-pounds.

U = Heat units in total steam used, in excess of heat units in equal weight of water at 212°.

u = (1st) Heat units returned from engine to boiler, plus (2d) heat units in steam used for any purpose other than running the engine, in excess of heat units in equal weight of water at 212°.

The duty of engine No. 2, as per trial of September 26th, 27th, is shown by the formula as follows:

Duty per 1,100 pounds steam made from and at 212°

$$= \frac{8{,}754{,}116{,}000 \times 1{,}062{,}600}{88{,}453 \times 1{,}000} = \mathbf{105{,}164{,}600 \text{ foot-pounds,}}$$

the same result as obtained by the specified formula.

In the above-mentioned trial the heat of the feed-water was increased 38.9° by the addition of the jacket-water, and the amount of heat returned to boiler from engine therefore equals 88,453 × 38.9 = 3,440,821.7 heat units. The net duty of engine per 1,100 pounds steam made from and at 212° then equals

$$\frac{8{,}754{,}116{,}000 \times 1{,}062{,}600}{88{,}453{,}000 - 3{,}440{,}821.7} = \mathbf{109{,}421{,}100 \text{ foot-pounds.}}$$

In case calorimeter tests of the steam had been made, "u" would have been ncreased by the addition of the heat units contained in steam used for the tests, in xcess of heat units in equal weight of water at 212°.

RECORD OF DUTY TRIALS OF TWO GASKILL HORIZONTAL COMPOUND PUMPING ENGINES, AT CHESTNUT HILL PUMPING STATION, BOSTON WATER WORKS.

		ENGINE No. 1.	ENGINE No. 2.
1.	Date of trials	Aug. 9–10, 1888.	Sept. 26–27, 1888.
2.	Time of beginning	10.22 A. M.	9.50 A. M.
3.	Duration	23 h. 51 m.	24 h. 11 m.
4.	Total revolutions	27,402.5	28,316
5.	Revolutions per minute	19.15	19.51
6.	Net area of pump plunger	3.3652 sq. ft.	3.3652 sq. ft.
7.	Length of stroke per revolution (3 ft. × 4)	12 ft.	12 ft.
8.	Grade of water in pump well	123.14 "	123.38 "
9.	Pressure in force main reduced to grade in ft. by mercury gauge	245.59 "	246.14 "
10.	Total lift in ft.	122.45 "	122.76 "
11.	Total lift reduced to pounds	52,972 lbs.	53,165 lbs.
12.	Total weight of water evaporated in boilers	86,788.5 "	88,453 "
13.	Equivalent weight of water evaporated from and at 212°	89,843 "	91,566.3 "
14.	Steam pressure in boilers	79.5 "	79.9 "

15. Duty of engines as developed by trials:

$$= \frac{\text{Work done in foot-pounds}}{\text{Equivalent water evaporated from } 212^\circ} \times 1{,}100$$

$$\text{No. } 1 = \frac{27{,}402.5 \times 3.3652 \times 12 \times 52.972 \times 144}{86{,}788.5 \times \frac{1000}{966}} \times 1{,}100 = \mathbf{103{,}347{,}500.}$$

$$\text{No. } 2 = \frac{28{,}316 \times 3.3652 \times 12 \times 53.165 \times 144}{88{,}453 \times \frac{1000}{966}} \times 1{,}100 = \mathbf{105{,}164{,}600.}$$

		ENGINE No. 1.	ENGINE No.
16.	Mean vacuum in condenser	26.3 ins.	26.53 in
17.	Mean atmospheric pressure by barometer	30.4 "	29.35 "
18.	Mean temperature of air in engine-room	79° F.	70.3° F.
19.	Mean temperature of external air	75.1° F.	60.8° F.
20.	Total volume of water pumped by plunger displacement, gallons	8,275,555	8,551,4
21.	Total volume of water pumped by weir-measurement, gallons	8,112,725	8,377,0
22.	Average slip of pumps	2%	[illegible]
23.	Indicated horse-power as determined by two sets of cards for each trial	210.7	214.4
24.	Horse-power in water lifted, pump-measurement, no allowance for slip	178.66	182.7
25.	Work done by pumps in per cent. of indicated horse-power	84.79%	85.2
26.	Temperature of feed-water from force main.	72.5	61.2
27.	Temperature of feed-water after receiving jacket-water	104.95	100.1
28.	Percentage of feed-water from jackets	13.0%	15.
29.	Amount of feed-water received from jackets	12,281 lbs.	14,004 lb

The boilers used in connection with the engines are of the horizontal, return tubular type, with external furnaces. They are 78 inches in diameter, 17 feet 11⅛ inches long between tube sheets, and each boiler contains 151 tubes, 3 inches outside diameter.

Shells of boilers are 7/16-inch thick steel, single-riveted in roundabout seams, and double-riveted in horizontal seams. Tube sheets are ½-inch thick steel. All rivet-holes were punched ⅛-inch small, and reamed to size required for ⅞-inch diameter rivets.

A feed-water heater is located in the smoke-flue between the boilers. It is made of 80 brass tubes, each 2⅛ inches diameter, and 15 feet long, supported in racks on a portable carriage.

Samples of each sheet of steel used in the boilers were tested for tensile strength, elastic limit, and elongation.

The steel was made by Carnegie, Phipps & Co., of Pittsburgh, Pa., and was inspected and tested by G. W. G. Ferris, Jr., C. E., of the same city.

The boilers were built by George Miles of Boston, and the flue feed-heater was made by the Atlantic Works of East Boston.

The designs for the boilers and flue-heater were furnished by the City Engineer.

RECORD OF TESTS OF TWO BOILERS AT CHESTNUT HILL PUMPING STATION, BOSTON WATER WORKS.

		BOILER No. 2.	BOILER No. 1.
1.	Date of trials	Aug. 9–10.	Sept. 26–27.
1a.	Time of beginning of trial	10.08 A. M.	9.30 A. M.
2.	Duration of trial	24 h. 5 m.	24 h. 36 m.
	DIMENSIONS AND PROPORTIONS.		
3.	Grate surface	38.5 sq. ft.	49 sq. ft.
3a.	Area of least draught	6 " "	6 " "
4.	Water-heating surface	2,164 " "	2,164 " "
5.	Superheating surface	7 " "	7 " "
5a.	Heating surface in feed-water heater	931 " "	931 " "
6.	Ratio of water-heating surface to grate surface	56:1	44:1
	AVERAGE PRESSURES.		
7.	Steam pressure in boiler by gauge	79.5 lbs.	79.9 lbs.
8.	Absolute steam pressure	94.2 lbs.	94.3 lbs.
9.	Atmospheric pressure by barometer	30.04 ins.	29.25 ins.
	AVERAGE TEMPERATURES.		
11.	External air	73° F.	56° F.
13.	*Steam	326.2° F.	*
14.	Escaping gases before passing feed-water heater,	358° F.	399.8° F.
14a.	Escaping gases after passing feed-water heater,	208.5° F.	256.5° F.
15.	Feed-water before passing heater	104.95° F.	100.1° F.
15a.	Feed-water after passing heater	153.65° F.	163.2° F.
15b.	Cochituate water from force main	72.4° F.	61.2° F.

*Steam thermometer proved to be incorrect, and was broken while being tested for correction.

	FUEL.	BOILER No. 2.	BOILER No. 1.
18.	Coal consumed	9,446 lbs.	9,266 lbs.
19.	Total refuse dry { No. 1, 486.5 lbs. No. 2, 900 lbs. }	9.5%	5.3%
20.	Total combustible (weight of coal, item 18; less refuse, item 19)	8,546 lbs.	8,779.5 lbs.
21.	Coal consumed per hour	392.2 "	376.7 "
22.	Combustible consumed per hour	354.9 "	356.9 "
	WATER.		
26.	Total weight of water pumped into boiler and apparently evaporated	86,788.5 "	88,453 "
26a.	Check on above measurement by meter measurement	84,848 "	87,606 "
26b.	Per cent. less by meter	2.2	1.0 "
26c.	Feed-water taken from force main by meter measurement	75,486 lbs.	74,831 "
28.	Equivalent water evaporated into dry steam from and at 212° F	99,459.6 "	101,897.9 "
	{ Including feed-water heater Excluding " " " }	95,121.2 "	96,060 "
29.	Equivalent total heat derived from fuel in British thermal units:		
	Including feed-water heater	96,074,870	98,350,891
	Excluding " " "	91,856,948	92,769,506
30.	Equivalent water evaporated into dry steam from and at 212° F. per hour:		
	Including feed-water heater	4,130 lbs.	4,142 lbs.
	Excluding " " "	3,950 "	3,905 "
	ECONOMIC EVAPORATION.		
31.	Water actually evaporated per lb. of coal, from actual pressure and temperature	9.19 lbs.	9.55 lbs.
32.	Equivalent water evaporated per lb. of coal from and at 212° F.:		
	Including feed-water heater	10.53 "	11.00 "
	Excluding " " "	10.07 "	10.37 "
33.	Equivalent water evaporated per lb. of combustible from and at 212° F.:		
	Including feed-water heater	11.64 "	11.61 "
	Excluding " " "	11.13 "	10.94 "
	COMMERCIAL EVAPORATION.		
34.	Equivalent water evaporated per lb. of coal, with one-sixth refuse at 70 lbs. gauge pressure, from temperature of 100° F., = item 33 multiplied by .7249 —		
	Including feed-water heater	8.44 "	8.42 "
	Excluding " " "	8.07 "	7.93 "

RATE OF COMBUSTION.

			BOILER No. 2.	BOILER No. 1.
35.	Coal actually burned per sq. ft. of grate surface per hour		10.19 lbs.	7.69 lbs.
36.	Consumption of dry coal per hour. Coal assumed with one-sixth refuse.	Per sq. ft. of grate surface	11.07 "	8.78 "
37.		Per sq. ft. water-heating surface	.197 "	.198 "
38.		Per sq. ft. of least draught	71.01 "	71.7 "

RATE OF EVAPORATION.

39.	Water evaporated from and at 212° F. per sq. ft. of water-heating surface per hour, excluding feed-water heater		1.88 "	1.80 "
40.	Water evaporated per hour from temperature of 100° F. into steam of 70 lbs. gauge pressure, excluding feed-water heater.	Per sq. ft. of grate surface	89.8 "	69.8 "
41.		Per sq. ft. of water-heating surface	1.59 "	1.57 "
42.		Per sq. ft. of least area for draught	578 "	566.8 "

COMMERCIAL HORSE-POWER.

43.	On the basis of 30 lbs. of water per hour evaporated from temperature of 100° F. into steam of 70 lbs. gauge pressure (= 34½ lbs. from and at 212°)			
	Including feed-water heater		119.7 h. p.	120.1 h. p.
	Excluding " " "		114.5 "	113.2 "

NOTE.—During a trial on August 22–23, Boiler No. 1 evaporated 10 lbs. of water per pound of coal, or an equivalent evaporation of 11.57 lbs. from and at 212°. The lower evaporation obtained in the trials recorded above are believed to be largely due to the inferior quality of coal used.

The logs of the tests are shown in the table.

LOG OF TEST OF ENGINE No. 1 AND BOILER No. 2 AT CHESTNUT HILL PUMPING STATION, BOSTON, AUG. 9-10, 1888.

			GAUGES.						TEMPERATURES.								
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
Time.	Engine counter.	Revolutions of engine.	Mercury gauge on force main of engine.	Float-gauge showing level of water in pump-well of engine.	Steam in boiler.	*Steam at engine.	Vacuum in condenser.	Atmospheric pressure by barometer near engine.	† Steam near boiler.	Feed-water before passing heater.	Feed-water after passing heater.	Escaping gases before passing feed-water heater.	Escaping gases after passing feed-water heater.	Air near engine.	External air.	Feed-water before receiving jacket-water.	Feed-water after receiving jacket-water.
Aug. 9.			Feet above W.W. base.	Feet above W.W. base.	Lbs.	Lbs.	Inches.	Inches.	Deg. F.	Deg. F.	Deg. F.	Deg. F.	Deg. F.	Deg. F.	Deg. F.	Deg. F.	Deg. F.
10.08 a.m.	595,792	...	238.70	123.71	68¾	68.9	.....	30.00	...	105½	160	...	...	81	Regular readings of thermometer not taken. Average obtained from a recording and a maximum and minimum thermometer.	....	...
10.22 "	595,792	146	244.48	122.66	77½	78.9	....	.	323	112	......	276	202	....		....	....
10.30 "	595,938	598	246.55	123.04	76½	76.9	26½	30.00	...	97½	189	...	...	81.5		...	80
11.00 "	596,536	596	245.60	123.14	79½	79.9	26¼	30.00	326	89	208	324	214	82		72	90
11.30 "	597,132	586	246.00	123.12	80	80.9	26¼	30.01	329	105½	156½	...	...	83		....	107
12.00 m.	597,718	574	245.00	123.18	80	80.4	26¼	30.00	327	104	143	324	204	83.5		72	103
0.30 p.m.	598,292	578	245.87	123.11	80	80.9	26¼	30.00	327	104	149	...	...	85		....	102
1.00 "	598,870	587	245.12	123.18	79½	80.4	26¼	30.00	326	105	141	322	194	84.5		73	104
1.30 "	599,457	587	246.25	123.07	80	81.4	26¼	30.00	328	102	146	..	...	85		...	105
2.00 "	600,044	576	245.50	123.15	79½	79.9	26¼	29.98	327	110	155	334	197	85.5		72½	111
2.30 "	600,620	580	245.60	123.10	80	81.4	26¼	29.97	326	114½	156	..	..	86.5		...	111
3.00 "	601,200	580	246.00	123.10	79½	79.9	26¼	29.96	327	99	154½	366	207	88		73	98
3.30 "	601,780	578	246.25	123.06	80	80.9	26¼	29.98	326	104	157	..	..	87		..	104
4.00 "	602,358	571	245.25	123.12	80	80 4	26¼	29.98	326	108	153	364	208	87		72½	109
4.30 "	602,929	572	245.37	123.10	79½	80.4	26¼	29.98	326	107	160½	...	...	87		....	108
5.00 "	603,501	570	245.10	123.12	79½	79.9	26¼	29.98	326	110	166	370	214	87.5		72½	110
5.30 "	604,071	574	245.40	123.12	80	80.4	26¼	29.98	326	110	162	..	...	87		....	109
6.00 "	604,645	579	246.00	123.12	79½	79.9	26¼	29.98	326	102	151	360	214	87		72½	99
6.30 "	605,224	571	246.00	123.10	79½	79.9	26¼	29.98	326	102	150½	...	...	85		....	101
7.00 "	605,795	563	245.20	123.12	79½	79.9	26¼	29.98	326	103	146	362	213	83		72½	103
7.30 "	606,358	571	245.75	123.09	80	81.4	26¼	30.00	326	103	152	...	...	....		...	104
8.00 "	606,929	567	245.40	123.16	80	80.4	26½	30.00	326	109	158	356	208	82		72	109
8.30 "	607,496	562	245.10	123.13	80	80.9	26.4	30.00	326	113½	159½	...	...	81		....	118
9.00 "	608,058	560	245.00	123.19	80	80.9	26.4	30.00	326	108	150	341	208	82		72½	108
9.30 "	608,618	569	245.75	123.16	79½	80.4	26.4	30.00	326	102	150	...	...	82		....	103
10.00 "	609,187	567	245.80	123.10	80	80.9	26.35	30.00	326	100½	141	348	206	81		72½	101
10.30 "	609,754	574	245.20	123.16	80	80.9	26.3	30.02	326	99½	146	...	...	80.5		....	103
11.00 "	610,328	565	246.00	123.15	79½	80.4	26.3	30.02	326	106	155	348	208	80		72½	108
11.30 "	610,893	559	245.35	123.14	80½	80.9	26.3	30.03	326	118	160	...	...	79		....	121
12.00 "	611,452	559	245.10	123.22	79½	79.9	26.3	30.04	326	115	154	348	206	78		72½	114
Aug. 10.																	
0.30 a.m.	612,011	560	245.50	123.18	80	80.4	26.3	30.04	326	103	158	..	...	77.5		....	102
1.00 "	612,571	564	245.40	123.15	80½	81.4	26.3	30.04	327	98	153½	366	210	76.5		72½	99
1.30 "	613,135	556	245.00	123.24	80	80.4	26.3	30.05	327	103½	144	..	...	76		...	104
2.00 "	613,691	551	245.25	123.20	80½	80.9	26.3	30.05	327	100	145	362	210	75		72½	104
2.30 "	614,242	552	245.33	123.18	80	80.9	26.3	30.06	327	107	149	..	...	73.5		...	109
3.00 "	614,794	569	245.00	123.22	80	80.9	26.3	30.07	327	107½	146	358	206	73.5		72½	109
3.30 "	615,363	571	246.50	123.13	79	79.9	26.3	30.07	328	107½	151	...	..	73		....	108
4.00 "	615,934	557	245.20	123.24	79½	79.9	26.3	30.08	326	105	153½	374	210	71.5		72½	104
4.30 "	616,491	573	245.00	123.26	80	80.9	26.3	30.10	327	101	153	...	...	70.5		....	106
5.00 "	617,064	595	245.75	123.20	79	79.9	26.3	30.11	327	104	148	374	210	70		72½	107
5.30 "	617,659	601	246.25	123.13	80	80.9	26.3	30.11	326	104½	149½	.	...	70		....	108
6.00 "	618,260	584	246.25	123.11	80	80.9	26.3	30.12	326	105	150	374	210	69		72½	107
6.30 "	618,844	586	245.87	123.10	80	81.4	26.3	30.14	326	107	148½		..	69.5		....	109
7.00 "	619,430	592	246.20	123.09	79½	80.4	26.3	30.14	326	103½	148	374	210	69.5		72½	106
7.30 "	620,022	590	245.40	123.16	79	79.9	26.25	30.13	326	103	148½	..	..	71		....	103
8.00 "	620,612	584	245.75	123.10	80	80.4	26.3	30.13	326	104	154½	376	208	72		72	105
8.30 "	621,196	580	245.75	123.11	79	79.9	26.3	30.13	326	108	147½	...	..	72		..	104
9.00 "	621,776	578	245.30	123.14	79	79.9	26.3	30.13	325	105	151½	382	212	73		72	107
9.30 "	622,354	583	246.40	123.07	79½	80 9	26.3	30.13	326	99½	151½	...	...	75.5			96
10.00 "	622,937	257.5	245.60	123.06	75½	76.9	26.3	30.15	324	107½	157	386	216	76		72	110
10.30 "	623,194.5	...	245.60	123.40	68½	68.9	.....	30.15	...	110½	162	...	...	76			......
....	....	27,402.5	245.50	123.14	79½	79.9	26.3	30.04	326.2	104.95	153.05	358	204.5	79.0	73	72.4	105.2

* This steam gauge apparently reads too high.

† This thermometer apparently reads too high.

LOG OF TEST OF ENGINE No. 1 AND BOILER No. 2 AT CHESTNUT HILL PUMPING STATION, BOSTON, AUG. 9-10, 1888.—*Continued.*

			QUANTITY OF WATER FED TO BOILER.									WATER EVAPORATED.		
			Meter Measurement used as Check on Weighing.				By Weighing.							
1	19	20	21	22	23	24	25	26	27	28	29	30	31	32
Time.	Coal fed to boiler, including 40 per cent. of wood.	Amount of steam used by calorimeter.	Reading of feed-water meter.	Cubic feet per 30 minutes.	Pounds per 30 minutes.	Total amount fed to boiler from the beginning of the test.	Amount of cold feed-water.	Amount of water from steam-jackets.	Quantity weighed and emptied into feed-tub.	Quantity fed to boiler per 30 minutes, corrected meter readings.	Total amount fed to boiler from the beginning of the test.	Height of water in boiler above bottom of glass gauge.	Quantity of water in boiler above assumed datum.	Evaporation per 30 minutes.
Aug. 9.	Lbs.	Lbs.	Cubic Feet.	Cubic feet.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Inches.	Lbs.	Lbs.
10.08 a.m.	......	See report on Calorimeter Test.	67,707½	....	.....	....	.....	Total amount = column 27 less column 25.	......	.....	......	4⅞	1,030	.....
10.22 “	......		......	3.5	217	217	249		......	217	217	...	.....	217
10.30 “	......		67,711	31	1,920	2,137	2,367		2,302.5	1,963	2,180	...	.	1,963
11.00 “	......		67,742	25	1,548	3,685	810		1,281	1,579	3,759	4½	875	1,734
11.30 “	......		67,767	33	2,044	5,729	1,619		2,262	2,094	5,853	3¾	540	2,429
12.00 m.	......		67,800	32	1,982	7,711	2,553		2,164	2,032	7,885	4⅛	710	1,862
0.30 p.m.	......		67,832	34	2,106	9,817	1,557		1,737.5	2,156	10,041	4½	875	1,991
1.00 “	......		67,866	26	1,610	11,427	1,619		2,281.5	1,641	11,682	5	1,075	1,441
1.30 “	......		67,892	20	1,239	12,666	810		649.5	1,264	12,946	4½	875	1,464
2.00 “	......		67,912	30	1,858	14,524	1,432		2,267	1,895	14,841	3 5/16	325	2,445
2.30 “	......		67,942	39	2,415	16,939	2,803		2,461	2,471	17,312	3½	425	2,871
3.00 “	......		67,981	28	1,734	18,673	1,557		1,902.5	1,771	19,083	4⅝	925	1,271
3.30 “	......		68,009	28	1,734	20,407	1,246		1,386	1,771	20,854	4¼	765	1,931
4.00 “	2,685.65		68,037	27	1,672	22,079	1,744		2,114.5	1,709	22,563	4⅛	710	1,764
4.30 “	......		68,064	26	1,610	23,689	1,931		1,489.5	1,648	24,211	4	650	1,708
5.00 “	......		68,090	28	1,734	25,423	747		1,604.5	1,771	25,982	3½	425	1,996
5.30 “	......		68,118	39	2,415	27,838	2,304		2,487	2,477	28,459	3 5/16	325	2,577
6.00 “	......		68,157	34	2,106	29,944	1,744		2,502	2,156	30,615	4 5/16	790	1,691
6.30 “	......		68,191	33	2,044	31,988	1,868		1,592.5	2,094	32,709	5⅛	1,125	1,759
7.00 “	......		68,224	24	1,486	33,474	1,557		1,966.5	1,517	34,226	5¾	1,370	1,272
7.30 “	......		68,248	21	1,301	34,775	1,121		1,235	1,322	35,558	5⅛	1,125	1,577
8.00 “	......		68,269	20	1,239	36,014	1,059		1,326	1,264	36,822	4⅛	710	1,679
8.30 “	......		68,289	36	2,230	38,244	2,429		2,054	2,285	39,107	3	145	2,850
9.00 “	2,344.75		68,325	34	2,106	40,350	1,308		2,162.5	2,155	41,262	4¼	765	1,535
9.30 “	......		68,359	35	2,168	42,518	2,491		2,106	2,224	43,486	5	1,075	1,914
10.00 “	......		68,394	27	1,672	44,190	872		2,086	1,709	45,195	6	1,470	1,314
10.30 “	......		68,421	16	991	45,181	623		541	1,016	46,211	5¾	1,370	1,116
11.00 “	......		68,437	20	1,239	46,420	1,059		1,405	1,264	47,475	4	650	1,984
11.30 “	......		68,457	28	1,734	48,154	1,682		1,898	1,771	49,246	3	145	2,276
12.00 “	......		68,485	38	2,353	50,507	1,931		2,530	2,409	51,655	3	145	2,409
Aug. 10.														
0.30 a.m.	......		68,523	27	1,672	52,179	1,059		1,446	1,709	53,364	4¼	765	1,089
1.00 “	......		68,550	34	2,106	54,285	2,553		2,213.5	2,155	55,519	5	1,075	1,845
1.30 “	......		68,584	26	1,610	55,895	1,495		1,985	1,648	57,167	4¾	975	1,748
2.00 “	......		68,610	27	1,672	57,567	1,370		1,408.5	1,709	58,876	5	1,075	1,609
2.30 “	......		68,637	26	1,610	59,177	1,183		1,445.5	1,648	60,524	4¾	975	1,748
3.00 “	......		68,663	26	1,610	60,787	1,432		1,602	1,647	62,171	4¾	975	1,647
3.30 “	2,266.25		68,689	30	1,858	62,645	1,744		2,394	1,902	64,073	4½	875	2,002
4.00 “	......		68,719	31	1,920	64,565	810		2,031.5	1,963	66,036	4⅝	925	1,913
4.30 “	......		68,750	29	1,796	66,361	2,304		1,765	1,833	67,869	5	1,075	1,683
5.00 “	......		68,779	28	1,734	68,095	1,931		1,609.5	1,771	69,640	5⅛	1,125	1,721
5.30 “	......		68,807	28	1,734	69,829	1,246		1,479.5	1,771	71,411	4¾	975	1,921
6.00 “	......		68,835	30	1,858	71,687	1,619		1,846	1,901	73,312	4¾	975	1,901
6.30 “	......		68,865	32	1,982	73,669	2,180		2,609.5	2,032	75,344	4¾	975	2,032
7.00 “	......		68,897	30	1,858	75,527	1,495		1,672	1,901	77,245	5⅛	1,125	1,751
7.30 “	......		68,927	22	1,363	76,890	1,183		1,363	1,393	78,638	5	1,075	1,443
8.00 “	......		68,949	37	2,291	79,181	1,869		2,584.5	2,347	80,985	4 3/16	735	2,687
8.30 “	......		68,986	23	1,424	80,605	1,059		858	1,455.5	82,440.5	5 5/16	1,200	990.5
9.00 “	......		69,009	35	2,168	82,773	2,367		2,630	2,223	84,663.5	4 5/16	785	2,688
9.30 “	......		69,044	21	1,301	84,074	1,121		1,451	1,332	85,995.5	4⅞	1,030	1,037
10.00 “	2,149.35		69,065	12.5	774	84,848	374		600	793	86,788.5	4⅞	1,030	793
10.30 “	......		69,077½	....	.....	.....	.....		......	.....	......	...	.....	...
......	9,446.0	1,369.5	......	1,370.	84,848	......	75,486	11,302.5	86,788.5	86,788.5	......	...	.....	86,788.5

LOG OF TEST OF ENGINE No. 2 AND BOILER No. 1 AT CHESTNUT HILL PUMPING STATION, BOSTON, SEPT. 26-27, 1888.

			GAUGES.						TEMPERATURES.								
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
Time.	Engine counter.	Revolutions of engine.	Mercury gauge on force main of engine.	Float-gauge showing level of water in pump-well of engine.	Steam in boiler.	Steam at engine.	Vacuum in condenser.	Atmospheric pressure by barometer near engine.	Steam near boiler.	Feed-water before passing heater.	Feed-water after passing heater.	Escaping gases before passing feed-water heater.	Escaping gases after passing feed-water heater.	Air near engine.	External air.	Feed-water before receiving jacket-water.	Feed-water after receiving jacket-water.
Sept. 26.			Feet above W.W. base.	Feet above W.W. base.	Lbs.	Lbs.	Inches.	Inches.	Deg. F.	Deg. F.	Deg. F.	Deg. F.	Deg. F.	Deg. F.	Deg. F.	Deg. F.	Deg. F.
9.50 a.m.	....	195	.....	.....	.....	.....	.....	.....	Thermometer incorrect, and was broken while being tested for a correction after this test.	...	...	...	...	....	Regular readings of thermometer not taken. Average temperature obtained from a recording and maximum and minimum thermometer.	....	...
10.00 "	195	598	246.25	123.04	80.3	79.5	26.6	29.15		88	181	334	228	69		62	103
10.30 "	793	572	247.25	123.05	80.3	80	26.5	29.13		101	204	...	...	68		....	104
11.00 "	1.365	575	246.40	123.48	80.3	80	26.3	29.08		100	158	324	212	67		61½	103
11.30 "	1,940	568	245.80	123.46	79.8	81	26.5	29.12		104	154	...	..	68		....	108
12.00 m.	2,508	592	246.25	123.42	80.3	80	26.5	29.11		105	150	314	210	67		61	104
0.30 p.m.	3,100	580	246.40	123.40	80.3	80	26.4	29.12		103	148	...	...	68		...	105
1.00 "	3,680	583	246.30	123.41	80.3	80	26.4	29.15		101	146	314	196	70½		61	100
1.30 "	4,263	594	246.40	123.41	80.3	80	26.3	29.18		102	142	...	..	71		..	106
2.00 "	4,857	577	246.30	123.41	80.3	80	26.5	29.25		98	140	318	196	71½		61	102
2.30 "	5,434	566	245.75	123.45	80.3	79.5	26.5	29.19		102	146	...	...	73		....	108
3.00 "	6,000	584	246.25	123.47	80.3	80	26.5	29.20		103	139	326	195	72		61½	98
3.30 "	6,584	589	246.10	123.39	79.8	79.5	26.5	29.23		94	146	...	...	72		....	94
4.00 "	7,173	591	246.20	123.39	79.8	79	26.5	29.25		96	150	340	208	71		61½	102
4.30 "	7,764	591	246.20	123.41	79.8	79	26.4	29.27		108	146	..	...	71		....	107
5.00 "	8,355	591	246.10	123.58	79.8	79.5	26.5	29.27		103	147	370	214	71		62	103
5.30 "	8,946	582	246.25	123.32	79.8	79	26.5	29.28		102	156	...	...	71		....	102
6.00 "	9,528	596	246.10	123.35	79.3	79	26.5	29.28		100	155	398	228	69		61	100
6.30 "	10 124	583	246.15	123.38	79.8	79	26.5	29.30		100	156	...	...	69		....	104
7.00 "	10,707	590	245.80	123.41	79.3	79	26.5	29.32		96	156	420	246	69		61½	93
7.30 "	11,297	586	246.00	123.41	79.8	79	26.5	29.34		99	164	...	...	69		..	108
8.00 "	11,883	588	245.75	123.44	79.3	79	26.5	29.35		98	153	430	258	69		61	98
8.30 "	12,471	579	245.80	123.44	79.3	79	26.5	29.37		96	166	...	...	69		....	101
9.00 "	13,050	583	245.75	123.43	79.3	79	26.5	29.37		98	168	436	274	69		61½	100
9.30 "	13,633	576	246.20	123.40	79.8	79	26.5	29.37		100	168	...	..	68		...	104
10.00 "	14,209	579	245.78	123.45	79.3	79	26.5	29.38		100	168	440	278	68		61	101
10.30 "	14,788	580	246.10	123.41	79.8	79.5	26.5	29.39		104	173	...	...	68		....	108
11.00 "	15,368	588	246.40	123.39	79.8	79	26.5	29.40		105	174	440	280	71		61	105
11.30 "	15,956	580	246.10	123.40	79.8	79	26.75	29.43		104	173	...	...	72		....	104
12.00 "	16,536	588	245.60	123.48	78.8	78	26.5	29.42		99	167	446	288	72½		61	102
Sept. 27.																	
0.30 a.m.	17,124	589	246.60	123.40	80.3	79	26.5	29.42		96	172	...	...	72		....	98
1.00 "	17,713	598	246.60	123.38	80.3	79	26.5	29.43		98	179	448	298	72		61	98
1.30 "	18,311	587	246.25	123.41	79.8	79	26.5	29.44		104	179	...	...	72		...	107
2.00 "	18,898	589	246.25	123.41	79.8	79	26.5	29.44		104	180	450	308	73		61	102
2.30 "	19,487	586	246.10	123.44	79.8	79	26.75	..		102	181	...	...	...		...	104
3.00 "	20,073	586	246.50	123.40	79.8	79	26.75	29.45		99	174	450	308	73		61	101
3.30 "	20,659	573	245.90	123.45	79.8	79	26.75	....		98	170	...	...	....		...	101
4.00 "	21,232	572	246.25	123.44	80.3	79.5	26.75	29.46		96	172	440	300	72½		61	100
4.30 "	21,804	591	245.75	123.49	79.8	79	26.75	...		104	172	...	...	...		...	106
5.00 "	22,395	591	246.20	123.42	79.8	79.5	26.5	29.48		100	162	430	294	72		61	102
5.30 "	22,986	588	246.00	123.39	80.3	79	26.5	....		99	167	...	...	...		...	100
6.00 "	23,574	588	246.10	123.41	79.8	79	26.75	29.52		98	166	438	300	72		61	100
6.30 "	24,162	588	246.20	123.36	80.3	79.5	26.5	....		99	169	...	...	...		...	102
7.00 "	24,750	590	246.25	123.35	79.8	79	26.5	29.50		102	166	438	294	68		61	104
7.30 "	25,340	582	246.00	123.33	80.3	79.5	26.6	...		102	169	...	...	...		...	105
8.00 "	25,922	580	246.25	123.28	79.8	79.5	26.75	29.52		104	168	428	290	69		61	102
8.30 "	26,502	586	246.15	123.28	79.8	79	26.5	..		99	164	...	..	...		...	100
9.00 "	27,088	597	246.00	123.27	79.3	79	26.5	29.53		100	166	424	290	71½		61	102
9.30 "	27,685	409	246.80	123.21	79.8	79	26.5	.....		102	171	..	...	...		...	104
9.50 "	28,094	222	246.50	123.25	79.8	77.5	26.5	...		...	...	...	...	..		....	...
10.01 "	28,316	...	......	......	...	...	.....	...		98	168	..		..		....	98
....	....	28.316	246.14	123.38	79.9	79.3	26.53	29.35		100.1	163.2	399.8	256.5	70.3	56	61.2	102.1

LOG OF TEST OF ENGINE No. 2 AND BOILER No. 1 AT CHESTNUT HILL PUMPING STATION, BOSTON, SEPT. 26–27, 1888.—*Continued.*

			QUANTITY OF WATER FED TO BOILER.									WATER EVAPORATED.		
			Meter Measurement used as Check on Weighing.				By Weighing.							
1	19	20	21	22	23	24	25	26	27	28	29	30	31	32
Time.	Coal fed to boiler, including 40 per cent. of wood.	Amount of steam used by calorimeter.	Reading of feed-water meter.	Cubic feet per 30 minutes.	Pounds per 30 minutes.	Total amount fed to boiler from the beginning of the test.	Amount of cold feed-water.	Amount of water from steam-jackets.	Quantity weighed and emptied into feed-tub.	Quantity fed to boiler per 30 minutes, corrected meter readings.	Total amount fed to boiler from the beginning of the test.	Height of water in boiler above bottom of glass gauge.	Quantity of water in boiler above assumed datum.	Evaporation per 30 minutes.
Sept. 26.	Lbs.	Lbs.	Cubic Feet.	Cubic feet.	Lbs.	Lbs.	Cubic feet.	Cubic feet.	Lbs.	Lbs.	Lbs.	Inches.	Lbs.	Lbs.
9.30 a.m.	....	None.	113,099	8	496	496	..	....	....	502	502	4⅛	710	502
10.00 "	....		113,107	31	1,922	2,418	..	....	2,135	1,940	2,442	...	....	1,940
10.30 "	....		113,138	27	1,674	4,092	50	16.7	2,165.5	1,698	4,135	4	650	1,753
11.00 "	....		113,165	34	2,108	6,200	..	....	855	2,133	6,268	4	650	2,133
11.30 "	....		113,199	18	1,116	7,316	44	12.6	2,438	1,128	7,396	3⅜	360	1,418
12.00 m.	....		113,217	28	1,736	9,052	..	....	1,647.5	1,755	9,151	3⅜	360	1,755
0.30 p.m.	....		113,245	30	1,860	10,912	51	7.6	1,981.5	1,878	11,029	3⅛	220	2,018
1.00 "	....		113,275	31	1,922	12,834	..	....	1,081.5	1,941	12,970	3¼	290	1,871
1.30 "	....		113,306	29	1,798	14,632	50	10.6	2,692	1,817	14,787	3½	425	1,682
2.00 "	....		113,335	21	1,302	15,934	..	....	880.5	1,314	16,101	3⅜	360	1,379
2.30 "	....		113,356	35	2,170	18,104	47	9.6	2,726.5	2,195	18,296	2½	0	2,555
3.00 "	....		113,391	35	2,170	20,274	..	....	1,986.5	2,195	20,491	3⅝	485	1,710
3.30 "	2,713.2		113,426	23	1,426	21,700	52	6.7	1,533.5	1,444	21,935	4	650	1,279
4.00 "	....		113,449	29	1,798	23,498	..	....	950	1,816	23,751	3⅛	220	2,246
4.30 "	....		113,478	29	1,798	25,296	46	12.6	2,701.5	1,817	25,568	3	145	1,892
5.00 "	....		113,507	30	1,860	27,156	..	....	1,017.5	1,878	27,446	3	145	1,878
5.30 "	....		113,537	29	1,798	28,954	51	8.6	2,065.5	1,817	29,263	3	145	1,817
6.00 "	....		113,566	31	1,922	30,876	..	....	1,956	1,941	31,204	3⅛	220	1,866
6.30 "	....		113,597	33	2,046	32,922	61	3.6	2,021	2,064	33,268	3¼	290	1,994
7.00 "	....		113,630	27	1,674	34,596	..	....	1,703	1,693	34,961	3½	425	1,558
7.30 "	....		113,657	36	2,232	36,828	47	16.7	2,177.5	2,256	37,217	3⅜	360	2,321
8.00 "	....		113,693	31	1,922	38,750	..	....	1,625.5	1,941	39,158	4	650	1,651
8.30 "	....		113,724	28	1,736	40,486	55	4.6	2,038.5	1,754	40,912	4	650	1,754
9.00 "	....		113,752	30	1,860	42,346	..	....	1,599.5	1,878	42,790	3⅞	595	1,933
9.30 "	2,130.5		113,782	27	1,674	44,020	48	9.6	1,929	1,693	44,483	4⅛	710	1,578
10.00 "	....		113,809	26	1,612	45,632	..	....	1,763.5	1,630	46,113	3⅞	595	1,745
10.30 "	....		113,835	27	1,674	47,306	41	12.6	1,511.5	1,693	47,806	3⅝	485	1,803
11.00 "	....		113,862	30	1,860	49,166	..	....	1,909.5	1,878	49,684	3¼	290	2,073
11.30 "	....		113,892	32	1,984	51,150	49	13.6	2,007.5	2,003	51,687	3¼	290	2,003
12.00 "	....		113,924	31	1,922	53,072	..	...	1,636.5	1,940	53,627	3¾	540	1,690
Sept. 27.														
0.30 a.m.	....		113,955	26	1,612	54,684	59	—1.4	2,017.5	1,621	55,258	4⅛	710	1,461
1.00 "	....		113,981	26	1,612	56,296	..	....	871.5	1,631	56,889	3¾	540	1,801
1.30 "	....		114,007	29	1,798	58,094	41	14.6	2,523	1,816	58,705	3¼	290	2,066
2.00 "	....		114,036	30	1,860	59,954	..	....	1,948	1,878	60,583	3⅛	220	1,948
2.30 "	....		114,066	31	1,922	61,876	50	11.6	1,907	1,941	62,524	3⅜	360	1,801
3.00 "	....		114,097	31	1,922	63,798	..	....	1,820	1,941	64,465	3⅝	485	1,816
3.30 "	2,274.5		114,128	25	1,550	65,348	53	8.6	1,721.5	1,568	66,033	3¾	540	1,513
4.00 "	....		114,153	30	1,860	67,208	..	....	1,952	1,878	67,911	3¼	290	2,128
4.30 "	....		114,183	31	1,922	69,130	48	13.6	1,848	1,941	69,852	3½	425	1,806
5.00 "	....		114,214	30	1,860	70,990	..	...	1,987	1,878	71,730	3⅝	485	1,818
5.30 "	....		114,244	32	1,984	72,974	55	7.6	1,955.5	2,003	73,733	3⅞	595	1,893
6.00 "	....		114,276	26	1,612	74,586	..	....	913.5	1,631	75,364	4⅛	710	1,516
6.30 "	....		114,302	28	1,736	76,322	49	5.6	2,504.5	1,755	77,119	3¾	540	1,925
7.00 "	....		114,330	29	1,798	78,120	..	...	957.5	1,817	78,936	3¾	540	1,817
7.30 "	....		114,359	30	1,860	79,980	47	12.6	2,718.5	1,878	80,814	3⅝	485	1,933
8.00 "	....		114,389	30	1,860	81,840	..	....	1,018	1,878	82,692	3¾	540	1,823
8.30 "	....		114,419	29	1,798	83,638	51	7.6	2,634	1,755	84,447	3¾	540	1,755
9.00 "	....		114,448	32	1,984	85,622	..	....	1,069	2,003	86,450	3⅝	485	2,058
9.30 "	2,148.0		114,480	32	1,984	87,606	55	9.6	2,230.5	2,003	88,453	5⅞	1,420	1,778
9.50 "	....		....	..	.....	......	..	....	525.5	.....	....	...	....	.....
10.01 "	....		114,512	...	.....	......	..	...	....	.....	....	4⅛	710	...
....	9,266.2		....	1,413	87,606	....	1,200	226.7	88,453	88,453	......	...	....	88,453

BOSTON, Aug. 30, 1888.

MR. WILLIAM JACKSON, *City Engineer, Boston* :

DEAR SIR—I beg to report as follows concerning my calorimeter test of the quality of the steam used by the Holly pumping engine at the Chestnut Hill Station of the Boston Water Works, during the duty trial of the same, made at your request on the 9th and 10th instant.

The calorimeter used for the test was one of my continuous superheating calorimeters, a description of which is given in the "Transactions of the American Society of Mechanical Engineers," Vol. VII., page 178, and Vol. VIII., page 235, and it was attached to the main 10-inch steam pipe at a point about midway between the boiler and the pumping engine. The sample of steam tested was drawn through a ½-inch perforated pipe which extended in a horizontal direction into the inside of the main, and diametrically across it.

The tests were in two series, one being made in the early part of the trial and one in the last part.

The steam used by the calorimeter was condensed in a surface condenser, and its weight determined. The whole quantity during the trial amounted to 1,369.5 pounds.

The accompanying record presents the full set of observations, taken after the apparatus had become heated to its normal state, corrected for errors of the thermometers ; and appended to the record are computations giving the percentage of moisture obtained from the average readings of the trial.

The result of the tests shows that the steam contained $\frac{24}{100}$ of one per cent. of moisture.

Respectfully submitted,

GEORGE H. BARRUS.

CALORIMETER TEST.

CHESTNUT HILL PUMPING STATION.

TIME.	PRESSURE.	TEMPERATURE OF STEAM SUPPLIED BY SUPERHEATER.		Temperature of Sample Steam when Discharged.	WEIGHT OF STEAM CONDENSED DURING PRECEDING HALF-HOUR.	
		Entering Jackets.	Leaving Jackets.		From Superheater.	From Sample.
Aug. 9.	Lbs.	Deg.	Deg.	Deg.	Lbs.	Lbs.
11.15 A. M.	80.	401.	372.	342.		
.30	79.6	401.5	372.5	343.	34.25	31.
.45	79.7	401.5	372.5	343.		
12.00	80.1	401.	372.	342.5	35.5	31.5
.15	80.	400.5	372.	342.5		
.30	79.4	402.5	373.5	343.	34.25	31.
.45	80.2	402.5	373.5	343.		
1.00	80.4	402.	373.	343.	35.5	31.5
.15	80.	402.5	373.5	343.		
.30	79.6	401.	372.5	343.	34.5	31.25
.45	79.7	403.	373.5	343.		
2.00	80.1	402.	373.5	343.5	34.25	30.75
.15	79.6	401.	373.	343.		
.30	79.7	403.	373.5	343.5	34.5	30.75
.45	79.1	403.	373.5	343.5		
3.00	80.1	402.5	373.5	343.	34.75	30.75
.15	79.7	402.5	373.5	343.		
.30	79.7	402.5	373.5	343.5	34.75	30.5
.45	79.9	403.5	374.	344.		
4.00	79.6	404.	375.	344.	34.75	30.5
.15	79.2	399.	372.	342.		
.30	79.9	397.	370.	342.	35.	30.5
.45	79.6	398.5	371.	342.		
5.00	79.2	398.5	371.	342.	35.	30.5
.15	79.6	399.	371.	342.		
.30	79.2	397.5	370.	341.5	35.	30.5
.45	79.2	398.	370.	341.5		
6.00	79.8	397.	370.	342.	34.75	30.75
.15	79.5	397.	370.	342.		
.30	79.8	398.	370.	342.	34.75	30.5
Aug. 10.						
8.15 A. M.	79.	397.	368.	340.		
.30	78.9	396.	368.	340.	35.	32.75
.45	79.6	396.5	369.	340.5		
8.00	79.	396.	368.	340.	34.75	33.
.15	79.2	395.	368.	340.		
.30	79.5	394.	367.5	340.	34.5	33.
verage and Totals	79.6	399.9	371.6	342.3	625.75	561.

CALORIMETER TESTS, CHESTNUT HILL PUMPING STATION.—COMPUTATIONS.

		Degrees.
1.	Superheat lost by jacket	28.3
2.	Superheat lost by jacket, corrected for 6 degrees of radiation	22.3
3.	Superheat lost by jacket, corrected for radiation and for ratio between steam used for superheater and that used for sample	24.7
4.	Superheat in sample steam when discharged	18.8
5.	Superheat in sample steam when discharged, corrected for 1.5 degrees of radiation from supply-pipe	20.3
6.	Superheat expended in evaporating the moisture contained in the sample leaving the main (line 3 minus line 5)	4.4
7.	Percentage of moisture (line 6 ÷ 18.7)	per cent. 0.24

WEIR MEASUREMENTS.

The following are extracts from the report on the weir measurements by Mr. Desmond FitzGerald, Resident Engineer and Superintendent Western Division Boston Water Works.

Description of Weir.

The 30-inch force main from the pumps delivers into one end of a brick compartment in the gate-house at Fisher Hill. The weir was built into the other end of the compartment in such a manner that the water could discharge freely into the air. The surface of the water in the reservior was maintained at a fixed height (grade 239.00), something over a foot below the crest of the weir, this being the height at which the correction for leakage from the chamber was made.

The elevation of the weir was 240.164. It was constructed free from end contractions. The sides consisted of a framework faced with boards planed smooth, free from irregularities, parallel and at right angles with the weir. The depth of the water for a considerable distance behind the weir was 20.24 feet, which gave an insignificant velocity of approach, and allowed the water to flow freely and smoothly to the weir. This action was still further aided by sets of screens placed near the delivery end of the force main. The lip was a steel straight edge, nickel plated, 0.0066 in thickness, carefully leveled, before and after each test, by a delicate bubble, made by Buff & Berger. The exact length was determined by a special brass rod fitted with sliding scale and vernier. The mean of a number of measurements made the length of the weir 5.008 feet.

Hook Gauge.

The hook gauge was arranged after the method recommended by Fteley and Stearns in their hydraulic experiments.

A plate was fitted normal to the current on the side of the box, at a distance of 6 feet above the weir and 0.4 feet below the crest. From the center of the plate a rubber hose led to the gauge-chamber, which was conveniently placed in another compartment of the gate-house adjoining the weir chamber, and separated therefrom by a water-tight dam.

The 0 of the gauge was placed level with the lip of the weir by means of the same apparatus used by Fteley and Stearns, and which gives a very delicate method of adjusting the levels. It is fully described and illustrated in the proceedings of the American Society of Civil Engineers.

OBSERVATIONS.

The heights of the water flowing over the weir were taken every five minutes throughout the duty trials by two observers, who alternated in descending into the chamber. Each observer checked the previous reading before taking a new observation. By this method errors were reduced to a minimum.

FORMULA.

In the general formula for flow over a weir with end contractions suppressed, the co-efficient 3.35 was adopted from the Fteley and Stearns experiments on a weir of same length and about the same flow. The correction for the velocity of approach was computed after the method recommended by Hunking and Hart, and was found to be 1.00043 for August 9–10, and 1.00044 for September 26–27. For five minutes, or three hundred seconds, we have

$$\Sigma\, Qg = 7.48 \times 300 \times 3.35 \times 5.008 \times 1.0004^3 \times (\Sigma\, H_m^{\frac{3}{2}})$$

The following method was pursued in working out the details. The heights at the beginnings and ends of the 5-mt. periods were added and their mean taken for a new height = H_m.

The $\frac{3}{2}$ power of this height was then found and placed in a column.

The total sum of the number in this column gave $(\Sigma\, H_m^{\frac{3}{2}},)$ completing the numerical values in the above formula.

This process is somewhat laborious, but is believed to give results that will be found trustworthy.

THE TESTS.

The following is a brief summary of the test made on Aug. 9–10, 1888. Water arrived at weir at 10h. 21m., August 9, and ceased running at 10h. 13m., August 10.

Treating first the even periods of 5 mts. we have $\Sigma\, H_m^{\frac{3}{2}} = 213.7545$

$\therefore\ \Sigma\, Qg = 7.48 \times 300 \times 3.35 \times 5.008 \times 1.00043 \times 213.7545 =$	8,050,700 gals.
Flow from 10h. 21m. to 10h. 25m., August 9	21,558 "
Flow from 10h. 10m. to 10h. 13m., August 10	16,964 "
Rise in weir box before normal flow	1,661 "
Leakage (measured), main, gates, weir, etc.	21,842 "
Total	8,112,725 gals.

The following extract from the log for one hour is sufficient to show the details, Aug. 9, 1888:

Time.	H.	H_m.	H_m^3.	$H_m^{\frac{3}{2}}$	$\Sigma\, H_m^{\frac{3}{2}}$
12–00	0.833	0.8360	0.5843	0.7644	13.9576
05	0.824	0.8285	0.5687	0.7541	
10	0.837	0.8305	0.5728	0.7568	
15	0.834	0.8355	0.5837	0.7637	
20	0.835	0.8345	0.5811	0.7623	
25	0.860	0.8475	0.6087	0.7802	
30	0.858	0.8590	0.6338	0.7961	
35	0.848	0.8530	0.6207	0.7878	
40	0.823	0.8355	0.5832	0.7637	
45	0.852	0.8375	0.5874	0.7664	
50	0.828	0.8400	0.5927	0.7699	
55	0.801	0.8145	0.5403	0.7351	
13–00	0.807	0.8040	0.5197	0.7209	9.2005

The test of September 26–27, 1888, was treated in a similar manner. It will be unnecessary to repeat the methods pursued. The following is a brief summary:

Water arrived at weir at 9 hours 49½ minutes, September 26th, and stopped 9 hours and 50 minutes, September 27th, giving a total flow of	8,353,497 gallons.
Add leakage and rise in weir box	23,503 gallons.
Total	**8,377,000 gallons.**

BOSTON, MASS.

OFFICIAL REPORT OF YEARLY DUTY

OF THE GASKILL PUMPING ENGINES AT CHESTNUT HILL STATION, SHOWING AN AVERAGE ON ACTUAL WATER PUMPED AND ACTUAL COAL CONSUMED OF 95,445,400 FOOT-POUNDS.

Office of City Engineer, City Hall,
Boston, February 17, 1890.

Mr. C. G. Hildreth, *Secretary Holly Company:*

Dear Sir—At Mr. Jackson's request, I send proof-sheet of work done at Chestnut Hill Station during the past year. The report will be ready for distribution in about ten days. The gallons pumped per revolution was taken at 295 instead of 302, the theoretical displacement, or an allowance of 2½ per cent. for slip. This is done to get the actual amount consumed in the district. The correction for steam used in heating was shown by a meter. The correction for steam used by dynamos was assumed to be thirty pounds of coal per hour, while the dynamo was in use—after calculation and experiments. You will notice that our boilers have done good work although banked 6½ hours per day.

Yours truly,
J. A. GOULD,
Assistant Engineer.

Note.—Adding the 2½ per cent. for slip, and counting the evaporation at 10, instead of 9.78, the showing would be 99,869,590 as the average duty for the year, which, considering that the fires were banked and the pump not run for "6½ hours per day," is not only a remarkably good yearly record for the engine, but reflects creditably on the management of the engineer.

STATEMENT OF OPERATIONS AT THE CHESTNUT HILL PUMPING STATION FOR 1889.

(CITY DOCUMENT NO. 32, PAGE 50.)

1889.	ENGINE NO. 1.			ENGINE NO. 2.			Total amount pumped.	Daily average amount pumped.	Total amount of coal consumed.	Daily average amount of coal consumed.	Total ashes and clinkers.	Per cent. ashes and clinkers.	Quantity pumped per lb. of coal. No correction for heating and lighting.	Quantity pumped per lb. of coal. Corrected for heating and lighting.	Average lift in feet.	DUTY IN FT.-LBS. PER 100 LBS. OF COAL.			Water evaporated in Boilers per lb. of coal.	
	Total pumping time.		Amount pumped.	Total pumping time.		Amount pumped.										Without correcti'n for heating and lighting.	Corrected for heating buildings.	Corrected for heating and lighting.	Actual.	From and at 212°.
Month.	Hrs.	Min.	Gallons.	Hrs.	Min.	Gallons.	Gallons.	Gallons.	Lbs.	Lbs.	Lbs.	Per cent.	Gallons	Gallons	Feet.	Ft.-Lbs.	Ft.-Lbs.	Ft.-Lbs.	Lbs.	Lbs.
Jan	373	10	120,360,000	162	10	53,757,850	174,117,850	5,616,700	214,705	6,926	16,047	7.5	811.0	873.4	122.78	83,041,400	89,435,600	89,435,600	9.47	11.23
Feb ..	510	15	170,303,500		..	....	170,303,500	6,082,300	198,564	7,092	14,428	7.3	857.7	933.2	123.28	88,182,400	95,942,500	95,942,500	9.72	11.52
March..	425	..	144,535,250	101	45	32,656,500	177,191,750	5,715,900	209,946	6,772	15,193	7.2	844.0	904.8	123.48	86,915,800	93,173,700	93,173,700	9.51	11.31
April...	....	..	....	511	50	162,530,250	162,530,250	5,417,700	189,852	6,328	13,517	7.1	856.1	898.0	122.92	87,762,200	92,062,500	92,062,500	9.67	11.52
May....	....	..	....	571	15	183,062,250	183,062,250	5,905,200	196,913	6,352	14,900	7.6	929.7	929.7	122.72	95,149,300	95,149,300	95,149,300	9.66	11.26
June ...	209	50	70,932,750	349	55	112,940,750	183,873,500	6,129,100	195,862	6,529	14,702	7.5	938.8	945.4	122.35	95,794,200	95,794,200	96,464,800	9.75	11.30
July....	563	..	187,811,750	....	..	....	187,811,750	6,058,400	207,265	6,686	15,309	7.4	906.1	945.1	122.46	92,545,900	92,545,900	96,525,400	9.85	11.36
August.	551	30	188,040,100	....	..	....	188,040,100	6,065,800	212,773	6,864	16,040	7.4	883.8	925.3	122.87	90,562,000	90,562,000	94,818,400	9.71	11.22
Sept ...	545	..	189,435,725	....	..	....	189,435,725	6,314,500	214,604	7,153	18,309	8.5	882.7	931.2	123.15	90,661,900	91,296,500	95,678,700	9.70	11.26
October	....	..	....	534	..	182,895,575	182,895,575	5,899,900	212,419	6,852	17,847	8.4	861.0	960.9	124.06	89,085,600	93,361,700	99,4[illegible]0,200	10.25	11.90
Nov ...	....	..	....	489	50	169,846,250	169,846,250	5,661,500	201,672	6,722	17,945	8.9	842.2	951.7	124.27	87,285,600	91,911,700	98,621,300	9.96	11.61
Dec	438	15	152,121,470	65	..	22,695,825	174,817,295	5,639,300	208,176	6,715	19,253	9.2	839.8	953.2	124.18	86,970,400	91,927,700	98,730,300	10.13	11.88
Totals and averages.	3616	..	1,223,540,545	2785	45	920,385,250	2,143,925,795	5,873,800	2,462,751	6,747	193,490	7.9	870.5	928.8	123.21	89,454,300	92,719,400	95,445,400	9.78	11.44

PORT HURON, MICH.

DECEMBER, 1888.

REPORT OF

HENRY BURTON, M. E., AND ASA R. COLE, M. E.

PORT HURON, MICH., December 21, 1888.

To the Honorable Board of Water Commissioners of the City of Port Huron:

GENTLEMEN—Under your instructions handed the undersigned on the 17th inst., to make examination and investigation of the pumping engines lately placed in the City Water Works of this city by the Holly Manufacturing Company of Lockport, N. Y., we herewith submit our findings and would respectfully report:

1st. That upon examination we find that the Holly Manufacturing Company have placed in the City Water Works one set of pumping engines of the horizontal beam, rotative type, compound, condensing, consisting of four cylinders, two high and two low-pressure, of the following dimensions: 18 inches and 36 inches by 32 inches, working in common on one shaft with fly-wheel between them. Each engine being connected to pumps of the following size: 22 inches in diameter and 32 inches in stroke.

We further find that the engine and pumps are constructed in a thorough and workmanlike manner, practically complying with the specifications.

2d. On the 19th inst. we commenced a capacity trial of six hours continued duration, running from 2.30 to 8.30 P. M., during which period the engines maintained a piston speed of $87\frac{1}{12}$ feet per minute under a steam pressure of seventy pounds, with a water pressure on the pumps of forty-five pounds. As the water is delivered by the pumps directly into the distributing mains its volume cannot be measured.

The capacity of the pumps is, therefore, found by computation from the displacement of the pump plungers.

Therefore we find that the engines made during this six hours run 5,895 revolutions, pumping 1,226,160 gallons under a water pressure of forty-five pounds, which equals **4,904,640 gallons** for twenty-four hours service of pump. You will notice that it only required a piston speed of $87\frac{1}{12}$ feet per minute to obtain the above result.

If a piston speed of 95 feet per minute were maintained the pumps would far exceed the amount, five million gallons, as specified in the contract.

he amount of coal consumed in making this six-hour run was 1,352 pounds, making of **102,638,623 pounds** per 100 pounds of coal [actual] consumed.

l. At the close of the six-hour test immediately followed an eighteen-hour test of stic duty, with boiler pressure at sixty pounds and a water pressure of forty-five ds.

'e find at the close of the test that there have been 2,358 pounds of coal consumed, umping 1,689,376 gallons [**less than half its nominal capacity**], making duty **l,750,886 foot-pounds** per hundred pounds of coal [actual] consumed.

h. At the expiration of the eighteen-hour test we made a half-hour pressure test, the following result: Boiler pressure seventy-five pounds, engine making nine lutions per minute, and maintaining a water pressure of from ninety to one hundred four pounds per square inch.

Very respectfully,

HENRY BURTON.
ASA R. COLE.

CHICAGO, ILLS.

Town of Lake Pumping Station, March, 1889.

REPORT OF BENZETTE WILLIAMS, C. E.

To the Honorable Board of Trustees of the Town of Lake, Illinois:

Gentlemen—In conformity to the directions contained in the letter of February 23, 1889, from your president, the Hon. Elmer Washburn, in which I was informed that I had been duly appointed to make a test of the new Holly pumping engines at your water works, I have the honor to report as follows:

Though it was the expressed desire of your Board that the test should be made at the earliest practicable day, I was not able to begin it until March 6th, owing to the contractors not being ready.

The contract with the Holly Manufacturing Company provides for the construction of one pair of Gaskill horizontal, compound, condensing crank and fly wheel pumping engines of 12,000,000 gallons daily capacity, and four tubular boilers.

The pair of engines as built is of the standard Gaskill type, consisting of two independent compound engines coupled to a common shaft, with cranks at right angles, said shaft carrying one fly-wheel seventeen feet in diameter and 40,000 pounds weight. The cylinders are steam-jacketed on heads and sides, the high-pressure cylinders of each engine being parallel with and directly above the corresponding low-pressure cylinder. The piston-rods being connected by short links to the opposite ends of a vertical walking-beam, gives the high and low-pressure pistons a travel in opposite directions.

The pump-plungers are double-acting and are in line with the low-pressure steam cylinders.

The high-pressure induction valves are double-beat poppets, so adjusted and operated by a drop cut-off as to be practically free from wire drawing.

The high-pressure eduction serve as the low-pressure induction-valves. They are gridiron slides worked by eccentrics from a horizontal shaft. The low-pressure eduction-valves are of the same kind and are operated by the same eccentric shafts.

The pump-valves are what are known as the "Troy Valve." They are rubber-faced and close quickly of their own weight and the pressure of the water. Being small and numerous, and opening but slightly, they allow but little loss of action.

The engines are provided with four vertical single-acting air-pumps and four vertical single-acting boiler feed-pumps operated from a horizontal rocking-shaft.

The exhaust steam is condensed with a jet condenser supplied with water through an independent suction-pipe connected with the pump-well.

Each pump is supplied by a 24-inch suction-pipe, which merges into a 30-inch pipe, common to both pumps, which connects with the pump-well at a distance of 33 feet horizontal and 15 feet vertical from the pumps.

The discharge main from each pump is 24 inches in diameter, and the two connect by a Y with a 30-inch main, which connects directly with the 24-inch force main and indirectly also with the 16-inch force main leading to the town, a distance of over three miles.

The main dimensions of the engine, as given by the specifications, are as follows:

Diameter high-pressure steam cylinders	27	inches.
Diameter low-pressure steam cylinders	54	inches.
Diameter pump-plungers	31	inches.
Stroke of steam-pistons and pump-plunger	40	inches.
Diameter of high-pressure piston-rods (one for each piston)	3½	inches.
Diameter of low-pressure piston-rods (two for each piston)	4½	inches.
Diameter of pump piston-rods	5	inches.

The boilers are of the horizontal return tubular type, with full, cast-iron fronts, made of Otis steel. They are brick set. The gases traverse a flue along the top of the boilers in reaching the chimney. They are fitted with one of Locke Bro.'s damper regulators.

Their dimensions are as follows:

Diameter of shell	66 inches.
Length of shell	18 feet.
Number of tubes	66
Diameter of tubes	4 inches.
Diameter of steam dome	33 inches.
Height of steam dome	44 inches.'
Water heating surface each boiler	1,440 square feet.
Length of grates (measured)	5 feet 10 inches.
Width of grates (measured)	5 feet 6 inches.
Area of grates each boiler	32 square feet.
Ratio of grate to heating surface	45
Length of flue from boilers to chimney	60 feet.
Area of flue from boilers to chimney	15 square feet.
Height of chimney	80 feet.
Area of chimney	20 square feet.

CAPACITY AND DUTY REQUIREMENTS.

By the terms of the contract the pumps are guaranteed to be capable of lifting 12,000,000 U. S. gallons of water in twenty-four hours against a head of 140 feet for domestic service, and against a head of 230 feet for fire service, at a piston speed not to exceed 114 feet per minute, and to perform a test duty of 95,000,000, foot-pounds with each 100 pounds of coal consumed. An evaporation of ten pounds of water to one pound of coal being assumed.

Steam pressure at boilers eighty pounds per square inch. The duty test to be made by three experts, at such reasonable time after starting the machinery on regular service as the contractors shall designate, and is to consist of a continuous run of twenty-four hours duration, and if the Board of Trustees shall so elect, after an interval of twenty-four hours, the engines are to be put upon another test of twenty-four hours, and the totals of these two runs of twenty-four hours each are to be the measure of arriving at the duty guarantee by the contractors.

The method of measuring the water pumped to be by weir measurement, or otherwise, as determined by the experts.

By agreement it was decided to leave the test to be conducted by one instead of three experts.

No evaporative test is required for the boilers, but by the terms of the contract three of the four boilers are to be capable of generating sufficient steam under eighty pounds pressure to the square inch to supply both engines while doing the maximum work specified.

It is further stipulated that as a whole the engines and boilers are to be the best of their kind as to material, workmanship, strength of parts and finish, and shall in all respects equal the pumping engine recently furnished the village of Hyde Park by the same contractors.

CONDITIONS OF TEST.

It was decided before beginning that two tests of twenty-four hours each should be made in accordance with the terms of the contract.

The first test was begun at 11 A. M. March 6th and ended at 11 A. M. March 7th, and the second test at 10.30 A. M. the 8th and ended at 1.30 A. M. the 9th inst.

Previous to this all necessary arrangements had been made to take the height of water in the pump-well, gauges had been tested, scales for weighing coal and feed water tested and sealed, and preparations made to take indicator cards from the four steam and two water cylinders at intervals of one hour.

On beginning each test the water was gauged in the boilers and the fires inspected, which at the close were brought as nearly as possible to the same condition. When unsuccessful in this the proper allowances have been made.

FEED WATER.

The apparatus provided for weighing the feed water consisted of a large cask resting on scales placed on an elevated platform, where the weighing was done, from whence the water was discharged into a lower tank with which the suction pipe of the feed pumps was connected.

It was arranged for the feed water to be taken direct from the force main, and for the water of condensation to be wasted.

After starting the first test, March 6th, it was found that the jacket water was not being wasted but was being trapped into the suction pipe of the feed pump without being accounted for. On objection being made to this arrangement, the contractors claimed that they are entitled to this advantage under the contract, and that in some other cases it has been allowed them.

It was concluded to continue the test as begun, and on the second test to provide for weighing the jacket water separately. As the amount of water condensed would be constant during equal periods of time, the jacket condensation in the second test has been added to the water weighed out during the first test in order to get the total amount of feed water.

The temperature of the feed water weighed was taken, but as the jacket water passed through a feed-water heater before being weighed there was no provision for determining the temperature of the feed water as it entered the boiler, except by assuming the jacket water to have the temperature due to fifty pounds pressure, and averaging it with the temperature of the water from the mains, which has been done 250 pounds has been added to the feed water of the second test to compensate for the inequality of water in the boilers at the beginning and end of the test.

CAPACITY.

As the town had to be supplied with water during the test, it was impossible to apply weir measurement to determine the water pumped; reliance has hence had to be placed upon the pump displacement, modified to produce the most probable results. The test of the Chicago pumps of the same capacity and design showed a loss of action averaging about three per cent. As the Chicago pumps worked during their test under about the same conditions of head, and as the loss of action was ascertained by careful weir measurement, it is not likely that the Town of Lake pumps will do better than they. I therefore consider it to be fair to the contractors to make a deduction of three per cent. for capacity and duty in this instance.

Previous to the trial, the pump plungers were measured by the town engineer, Mr. Annable, and found to be thirty-one inches in diameter. During the trial I measured the actual stroke of the pumps and found it to be 39¾ inches. These dimensions have hence been used in the calculations.

HEAD.

For both tests the gauges were uniformly read at equal periods of fifteen minutes, except that the water-pressure gauge was, for greater accuracy, read every five minutes during most of the second test. The greater duty shown in the table for the first test, is attributed to the inaccuracy of the gauge readings, owing to their infrequency and the irregular action of the engines, and is probably an error to the extent of the difference shown.

On the second test two gauges were placed on the gauge pipe and readings of both taken. The extra gauge had previously been tested and adjusted at the shops of J. P. Marsh, and before either test was begun, both the extra and the regular gauges were compared with a test gauge sent by Mr. Marsh to the works for that purpose. This comparison showed the extra gauge to register one pound and the regular gauge two and one-half pounds short of the test gauge, a difference of one and one-half pounds. The readings during the second test, however, when the gauges were under the influence of exactly the same conditions, after being corrected for difference in height, show a variation of only three-tenths of a pound, the extra gauge registering that much more than the regular gauge. The tendency of a gauge kept under pressure is to increase its readings, so that the relative change in the gauges can only be attributed to this cause. As the extra gauge had been carefully tested at the shop, and, as before said, the tendency of a gauge is to increase its reading, I have adopted the extra gauge as the standard and have in both tests added three-tenths of one pound to the mean reading of the regular gauge for the pressure. To the head due to this pressure has been added the height from the surface of the water in the pump-well to the gauge. Readings of the well-gauge were made every fifteen minutes.

In order to maintain a constant head and uniform speed of pumps, a part of the water pumped was allowed to escape through a blow-off back into the well.

INDICATOR CARDS.

Six indicators, one for each cylinder, were used, and cards taken every hour, making twenty-five sets in all for each test. The average pressures have been carefully ascertained from these cards by means of a planimeter, and the work done in the cylinders computed and a comparison made with the effective work, the results of which are given in Table No. 2.

The volume swept through by the piston during admission of the steam has also been ascertained from the cards, and the estimated weight of dry saturated steam computed, and compared with the steam actually supplied as shown by the feed water. These results are shown in Table No. 3.

On accompanying sheets are given exact copies of one set of indicator cards which are fair types of all.

BOILER RECORDS.

Youghioghaney—bituminous—coal was used during the test, that for the first coming from the Shoner mine and that for the second from the Nottingham mine. The difference in the evaporative power of the two coals as shown in the table, eight and seven-tenths pounds for the latter, and ninth and eight-tenths for the former, is very marked, and is only ascribable to a difference in the quality of the coals.

The coal from the Shoner mine burned without clinkering to a fine white ash. The Nottingham coal, on the other hand, clinkered very badly, and in so doing no doubt prevented a sufficient supply of air passing through the bed on the grates, producing incomplete combustion.

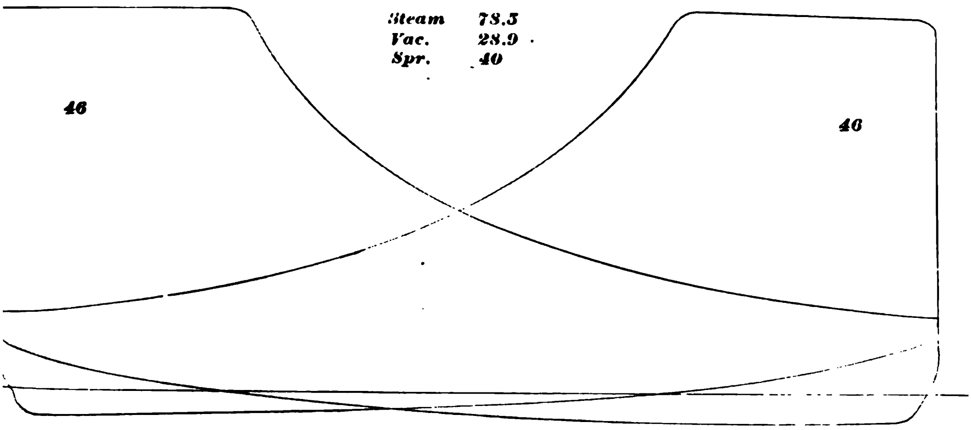
It is to be regretted that no observations as to priming were made, even though such observations are not essential to determine the fulfillment of the contract stipulations.

The boilers were fed during the whole of each trial with one feed pump.

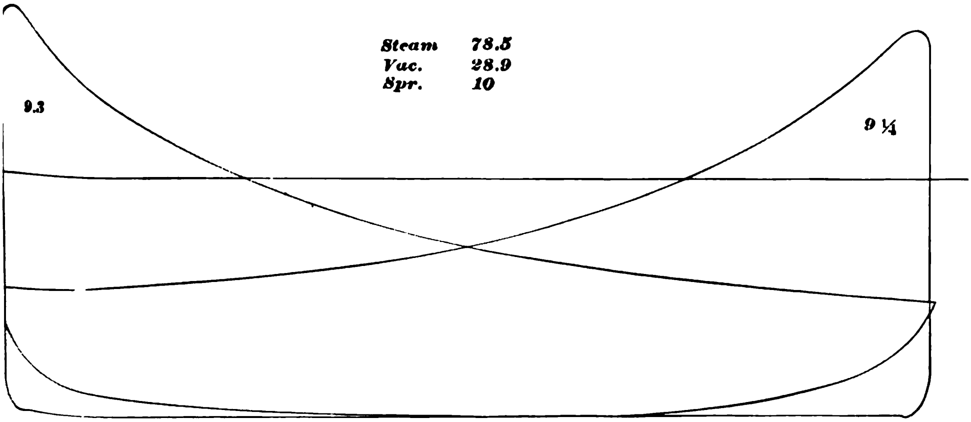
Indicator Cards, March 8th, 7.30 P. M.

EAST ENGINE.

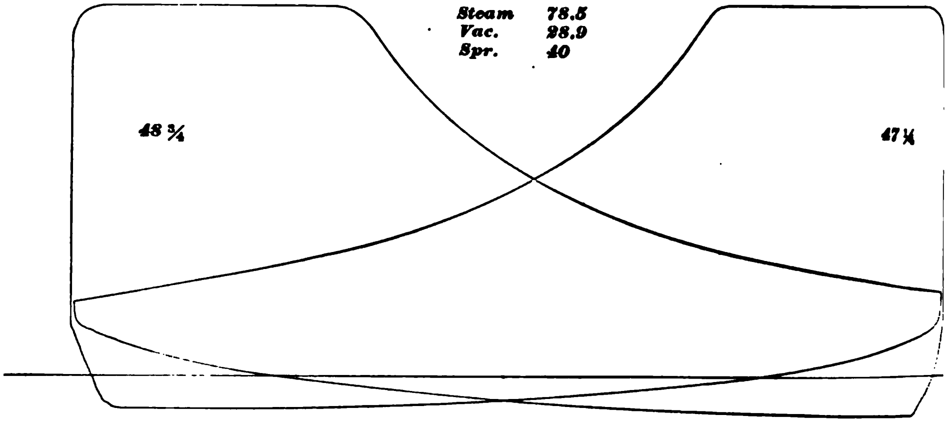
HIGH



LOW



HIGH


Steam 78.5
Vac. 28.9
Spr. 40
48 ¾
47 ¼

LOW

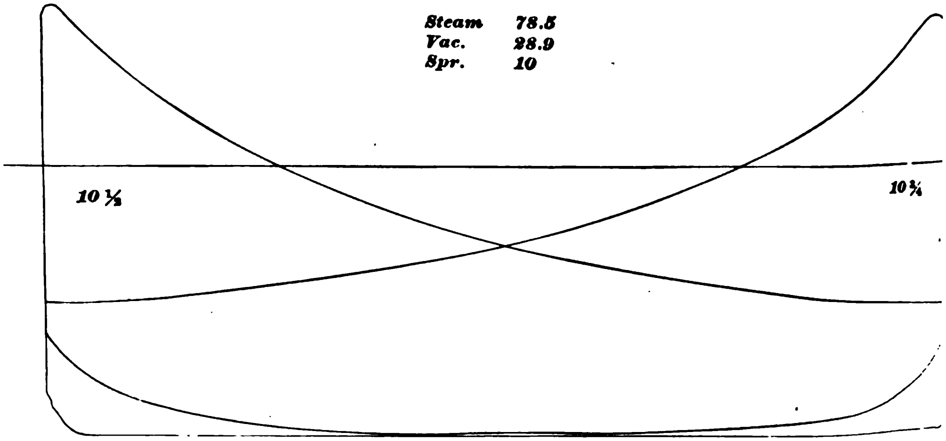

Steam 78.5
Vac. 28.9
Spr. 10
10 ½
10 ¾

TABLE No. 1.

GIVING THE AVERAGE RESULTS OF THE OBSERVATIONS TAKEN MODIFIED BY THE CORRECTIONS MADE.

	24 Hours Trial, 11 A. M. March 6 to 11 A.M. Mar. 7.	24 Hours Trial, 10.30 A. M. Mar. 8 to 10.30 A. M. Mar. 9.	Totals and Means, Both Trials.
Total revolutions	24,391	24,706	49,097
Revolutions per minute	16.94	17.16	17.05
Piston speed in ft. per minute (actual)	112.22	113.68	112.94
Piston speed for 12,000,000 gal. per day, (allowing 3 per cent. slip)	110.	....	....
Displacement gal. per revolution	517.08	....	....
Loss of action or slip (assumed)	3 per cent.	....	....
Water pumped, gal. per revolution	501.57	....	....
Water pumped, U. S. gal. in 24 hrs. (allowing 3 per cent. for slip)	**12,233,794**	**12,391,788**	**12,312,761**
Mean corrected pressure at gauge, pounds	52.8	54.90	53.85
Mean corrected pressure at gauge, ft. of water	121.90	126.84	124.37
Mean height from water in pump-well to gauge, feet	19.73	19.86	19.80
Mean total head against pumps, feet of water	141.63	146.70	144.17
Feed water taken from mains (pounds)	118,542	126,148	244,690
Feed water from jackets	13,648	13,648	27,296
Feed water, total amount used	132,190	139,796	271,986
Feed water, mean temperature (observed from mains 84°, calculated from jackets 29.8°)	58.6	57.5	58.
Coal consumed, pounds	16,686	19,618	36,304
Non-combustible, pounds	1,432	1,346	2,778
Combustibles, pounds	15,254	18,272	33,526
Duty foot-lbs. per 100 lbs. of coal, 10 lbs. of water evaporated being taken as the equivalent of 1 lb. of coal as per terms of contract	**109,054,000**	**108,200,000**	**108,000,000**
Duty foot-lbs. per lb. of coal (actual)	86,400,000	77,200,000	81,500,000
Duty foot-lbs. per lb. of combustible (actual)	94,500,000	82,800,000	88,200,000
Evaporation lbs. of water per lb. of coal	8.0	7.1	7.5
Equivalent evaporation from and at 212°	9.8	8.7	9.2
Coal consumed per sq. ft. grate surface per hr.	22.	25.5	23.6
Mean steam pressure at boilers	79.2	79.6	79.4
Mean steam pressure at engines	78.4	78.9	78.7
Mean steam pressure in jackets, approx	50.	50.	50.
Mean vacuum gauge, inches mercury	28.6	28.9	28.7
Temperature of engine-room	74.	....	....
Temperature of external air	....	26.	....

TABLE No. 2.

SHOWING THE MEAN RESULTS OF THE OPERATION OF THE ENGINES DURING THE TESTS, AS DERIVED FROM INDICATOR CARDS TAKEN EVERY HOUR, AND A COMPARISON OF THE INDICATED WORK WITH THE NET WORK.

	First Test 24 hours.	Second Test 24 hours.	Average for Two Tests.
Initial cylinder east engine, mean pressure of both ends	42.5	45.5	44.0
Initial cylinder west engine, mean pressure of both ends	44.0	46.4	45.2
Expansion cylinder east engine, mean pressure both ends	9.7	9.5	9.6
Expansion cylinder west engine, mean pressure both ends	9.5	10.8	10.25
Water cylinder east engine, mean pressure both ends	60.1	62.3	61.2
Water cylinder west engine, mean pressure both ends	57.8	60.7	59.25
Initial cylinders, total horse-power	166.7	179.6	173.15
Expansion cylinders, total horse-power	147.7	156.7	152.2
Total indicated horse-power of steam cylinders	314.4	336.3	325.4
Total indicated horse-power of water cylinders	297.0	313.0	305.0
Mean effective horse-power	310.5	319.8	315.1
Indicated H. P. of water cylinder per cent. of indicated H. P. steam cylinder	94.5	93.1	94.0
Efficiency, effective H. P. per cent. of indicated H. P.	98.7	94.8	96.9
Coal used per hour per ind. H. P. (contract basis) lbs.	1.75	1.73	1.74
Coal used per hour per indicated H. P (actual)	2.21	2.43	2.32
Steam used per hour per indicated H. P., lbs	17.5	17.3	17.4

TABLE No. 3.

SHOWING THE MEAN DISTANCE IN INCHES OF THE POINT OF CUT-OFF MEASURED FROM THE END OF THE INITIAL CYLINDERS OF BOTH ENGINES AS OBTAINED FROM INDICATOR CARDS, THE TOTAL POUNDS AND VOLUME OF STEAM AS DERIVED FROM THE ADMISSION VOLUME, AND A COMPARISON OF THE SAME WITH THE FEED WATER USED.

	First Test.	Second Test.	Both Tests
Mean point of cut-off, measuring from end of cylinder, inches	10.37	11.32	10.85
Clearance, inches	1.50	1.50	1.50
Mean equivalent length of admission, inches	11.67	12.92	12.35
Total volume swept through by pistons during admission, cubic feet	380,443	416,320	796,763
Equivalent pounds of dry saturated steam	81,993	90,112	172,105
Actual lbs. of steam as represented by the feed water	118,542	126,148	244,690
Per cent. of steam condensed during admission as shown	30.8	28.5	29.7

DISCUSSION OF RESULTS.—WORK.

The effective horse-power shown in Table No. 2 is greater than the indicated horse-power of the water cylinders. This is, of course, erroneous. The indicated horse-power of the pumps should be greater than the effective horse-power by one or two per cent., which represents the loss by the passage of the water through the suction pipes, pumps and valves to the point in the force main where the gauge pipe is attached. This discrepancy arises mainly, no doubt, from the inaccuracy of the indicator diagrams, caused by fluctuation of pressure, error in indicators, and the necessary inexactness in making up the average pressure from the cards which was done with a planimeter. It is also partly due to errors of pressure gauges and gauge readings, which in the case of the first test has a possible error of two per cent. Still another source of error is inequality of speed, and the impossibility of the mean observations being exact, owing to their infrequency.

The indicated horse-power of the water cylinders averages six per cent. less than the indicated horse-power of the steam cylinders. This is as it should be, and as the same causes of inaccuracy would operate in measuring the work in both the steam and water cylinders, the comparative results are doubtless about correct; hence six per cent. of the indicated power is shown to be used in overcoming the friction of the moving parts of the machine.

The effective horse-power as compared with the indicated horse-power of the steam cylinders is also too great, owing to the same causes which have produced the discrepancy between the effective horse-power and the indicated horse-power of the pumps.

As a whole, the results obtained of the indicated work verify the effective work as ascertained from the observations, showing conclusively that any errors that exist are small.

The cards from the steam cylinders indicate perfect working valves and an excellent distribution of the steam.

The clearance for the high pressure cylinders in terms of cylinder volume had previously been calculated at $3\frac{76}{100}$ per cent. This gives $1\frac{1}{2}$ inches in actual length of cylinder.

DUTY.

The causes operating to produce inaccuracy in the effective work, as above noted, will also affect the duty, but whether an elimination of these inaccuracies would increase or decrease the duty it is impossible to say. There is an undetermined element, however, which would affect the duty, and not the effective work. Whatever amount of priming may have occurred during the test would reduce the duty by increasing the feed water supplied. As no tests were made for entrained water the amount of this decrease cannot be known.

As the engines show an average of **108,600,000 foot-pounds** per 100 pounds of coal as **against 95,000,000** called for by the contract, the matter of slight inaccuracies is unimportant as far as they affect the acceptance of the machinery.

CAPACITY.

The pumps were guaranteed by the contractors to pump **12,000,000** gallons in 24 hours under a head of 140 feet. They delivered about *two per cent.* more than this quantity under fully *three per cent. more head.*

No test of the pumps was made with the fire pressure of 250 feet head. It would have been impracticable to have done this under the conditions of working without starting more pumps, and as these pumps worked so smoothly and noiselessly it was not considered necessary to apply this test.

INITIAL CONDENSATION.

Table No. 3 shows that there was apparently about 29.7 per cent. of the steam condensed during admission to the high pressure cylinder. Do doubt a small per cent. of this apparent condensation consisted of entrained water, but making allowance for this there would still be some 27 per cent. of the steam condensed on admission. This seems very high, but when we reflect that the jacket was 26 degrees less in temperature than the initial steam, and that the interior of the cylinders are cooled to a temperature due to about eight pounds pressure or 88 degrees less than the initial temperature, at the end of each stroke, we are not surprised.

The mean quantity of steam and entrained water admitted per stroke was 1¼ pounds or 27.7 cubic inches of water. Of this about 27 per cent. or 7½ cubic inches was condensed during admission.

The total mean area of the walls of the portion of the cylinders filled by initial steam at each stroke was about 2,200 square inches. A film of water covering this whole area, about $\frac{1}{300}$ of an inch in thickness, would equal 7.5 cubic inches. Thus it is seen how readily 27 per cent. of the steam may be condensed during admission.

BOILERS.

There being no evaporative test of the boilers required, no effort was made to secure a high record.

The evaporative performance shown should be equaled in ordinary daily working. The two west boilers of the four furnished by the contractors were used during the trial, and the results show them to be ample for the required work. So far as I was able to judge, the boilers have been well constructed and fill the contract in every respect.

In conclusion, I wish to say that the machinery as a whole I consider in every way up to, and in some way above, the contract requirements. The results of the test are entirely satisfactory to all concerned. The town and the manufacturers are to be congratulated upon the outcome as a whole.

I desire to express thanks to the contractors and the town officials for their kind assistance and courtesies.

Respectfully submitted.

BENZETTE WILLIAMS.

March 26th, 1889.

BUFFALO, N. Y.

MAY, 1889.

REPORT OF

LOUIS H. KNAPP, ENGINEER AND SUPERINTENDENT BUFFALO CITY WATER WORKS, AND FRANK W. HOLLY, ENGINEER AND SUPERINTENDENT HOLLY MANUFACTURING COMPANY.

BUFFALO, N. Y., May 25th and 26th. 1889

The machinery tested consisted of one Gaskill horizontal, compound pumping engine of twenty million gallons guaranteed capacity per twenty-four hours, and four horizontal return flue boilers, five and one-half feet diameter by eighteen feet long, manufactured by the Holly Manufacturing Company of Lockport, N. Y.

The following are the principal dimensions of the engine:

Diameter of high-pressure cylinders	36 inches.
Diameter of low-pressure cylinders	72 inches
Diameter of high-pressure piston rods	5 inches.
Diameter of low-pressure piston rods	5½ inches.
Stroke of each piston	48 inches.
Clearance in terms of cylinder, volumes H. P. cylinder	3 per cent
Clearance in terms of cylinder, volumes L. P. cylinder	2 5 per cent.
Diameter of pump plungers	36 inches.
Stroke of plungers	48 inches.
Diameter of plunger rods	6 inches.

DIMENSIONS OF BOILERS.

Diameter of shell	66 inches.
Length of shell	18 feet.
Number of tubes	80
Diameter of tubes	3½ inches.
Length of tubes	18 feet.
Grate surface, square feet	34

Heating surface in boiler, square feet	1502
Ratio of heating surface to grate surface	44.18
Chimney (8′ × 8′ square) square feet	64
Chimney, height	125

The duty trial commenced at 5.30 P. M., May 25th, and ended at 5.30 A. M., May 26th, 1889.

The contract requires that the engine and boilers as a plant complete shall develop a duty of one hundred and five million foot-pounds for every one hundred pounds of coal consumed under the boilers, and shall be computed by the following formula:

$$D = \frac{200 \times A \times P \times N \times S}{C} \text{ in which}$$

D = Duty in foot-pounds per one hundred pounds of coal.
A = Aggregate mean area of pump plungers.
P = Pressure per square inch of pump plungers.
N = Number of revolutions or double strokes of engine.
S = Stroke of both pump plungers.
C = Pounds of coal consumed under the boilers.

During the trial for duty, at intervals of fifteen minutes, readings were taken in the engine room of the revolution counter, steam and water-pressure gauges, vacuum gauge, suction-well gauge, temperature of air pump discharge or feed water in hot-well before passing through heater, and temperature of feed water after passing through heater and before entering boilers, and in the boiler room of the steam-pressure gauges, and height of water in boilers. An accurate account of the amount of coal burned under the boilers was also kept, it being weighed in iron wheelbarrows on new scales tested just before the beginning of the trial.

The diameter of plungers was measured and found to be 36″ according to the contract. The length of stroke was also determined by measurements to be 48 inches.

The coal used was a excellent quality anthracite, known as the "Henry Mine" coal.

As a result of the observations recorded in the engine and boiler rooms, the following averages were obtained:

Average water pressure by gauge	80.265 pounds.
Average height of center of water-pressure gauge above water in suction well	25.805 feet.
Average steam pressure on gauge at engine	80.265 pounds.
From which should be deducted the pressure due to the height of water column on the gauge	2. pounds.
Leaving an average of	78.265 pounds.
as the average steam pressure at the engine.	
Average vacuum	26.5 inches.
Average temperature of feed water	113.51° F.
Average temperature of air pump discharge, from which feed water to boilers was drawn	103.85° F.
Average increase in temperature of feed water by the addition of jacket water	9.66° F.
Average steam pressure at boilers	80.19 pounds.
The average head pumped against by the engine during the trial for duty, was, by the terms of the contract, the total head pumped against plus one pound for friction in the suction	92.435 pounds.
Number of revolutions of the engine during the trial	11,365
Stroke of pump plungers	4 feet

Then, $A = 2007.482$, $P = 92.435$, $N = 11{,}365$, $S = 4$, $C = 13{,}800$ pounds of coal burned during the trial.

$$D = \frac{200 \times 2007.482 \times 92.435 \times 11365 \times 4}{13800} = \mathbf{122{,}255{,}512.2}$$

foot pounds per 100 pounds of coal consumed under the boilers ; no allowance being made for ashes or any other purposes ; the engine pumping its own feed water by means of feed pumps attached directly to the engine ; no extra boilers being fired for this purpose. All fuel consumed during the trial was charged against the engine.

From time to time, during the trial for duty, indicator cards were taken from the steam cylinders and pumps. These cards were carefully calculated and the following results deducted therefrom :

Average water pressure as shown by pump indicator cards = 94.773 pounds per square inch, or 2.338 pounds more than the head as calculated by the terms of the contract.

Work in pump cylinders was 2.53% in excess of work allowed by terms of contract.

Duty from pump cards = **125,248,575.63 foot-pounds.**

The indicated work in the steam cylinders shows an excess of 7.23% over the indicated work in the pump cylinders, indicating that 93.16% of the work performed by the steam cylinders was transmitted to the pumps, and that 6.84% was used in driving air and feed pumps, overcoming friction and other losses in the engine.

Duty of engine calculated from steam indicator cards = **134,304,047.65** foot pounds per 100 pounds of coal.

Indicated horse-power of engine during duty trial.....	780 H. P.
Pounds of coal consumed per hour during duty trial.......................	1,150.
Pounds of coal consumed per hour per indicated horse-power...............	1.474.

The firing of the boilers was performed by the regular firemen, employed by the city who happened to be on duty during the test ; a change in firemen occurring at 11 o'clock P. M. May 25th, so that two different firemen were supplying coal to the furnaces during the test. Shortly after 11 o'clock, or on the arrival of the second fireman, the fires were cleaned, and 534 pounds of ashes weighed, as the result of the cleaning. At this time 6,400 pounds of coal had been fed into the furnaces, which would indicate that about 8.34% of the coal was ashes and other non-combustible substances.

Duty per 100 pounds of combustible, head on pumps calculated by contract requirements	**133,379,349.99**	**foot-pounds.**
Indicated duty of engine (from steam cylinder cards) per 100 pounds of combustible..................	**146,524,162.83**	**foot-pounds.**
Duty developed by pumping plant on test as calculated by contract requirements..................	**122,255,512.2**	**foot-pounds.**
Duty guaranteed by contract.......................	**105,000,000**	**foot-pounds.**
Excess over contract...........	**17,255,512.2**	**foot-pounds.**

or **16.43 per cent.** higher duty than guaranteed by the Holly Manufacturing Company.

Directly after the termination of the duty trial the test for capacity was begun, ending at 7.30 A. M., May 26th.

The contract formula for capacity is as follows

$Q = 145.122 \times A \times N \times S.$
A = Sum of effective area of both plungers.
N = Mean number of revolutions per minute.
S = Stroke of pumps in feet.

In the foregoing formula three per cent. is allowed for the possible imperfect filling of the pumps.

Readings were taken every fifteen minutes and the following averages obtained for the time beginning at 6 o'clock A. M., and ending at 7.30 A. M., May 26th, 1889.

Average head pumped against, calculated by contract requirements, pounds....	95.45
Average revolutions per minute..	18.1

$Q = 145.122 \times 2007.482 \times 18.1 \times 4 =$ **21,092,277.723 U. S. gallons** per 24 hours.

Figured from pump plunger displacement the capacity would be **21,744,616.209 U. S. gallons** per 24 hours.

Indicated horse-power of the engine during the capacity test = 918.5.

At the end of the duty trial the fires under the boilers were viewed, and found to be in as good condition as at the commencement of the test.

At the end of the capacity test all of the bearings were examined and found to be running perfectly cool with no signs of heating.

LOUIS H. KNAPP,
Superintendent and Engineer Buffalo Water Works.
FRANK W. HOLLY,
Engineer and Superintendent Holly Manufacturing Co.

SPRINGFIELD, OHIO.

August, 1889.

REPORT OF CHARLES A. BAUER, C. E.

[The following report is of the test of the Gaskill Horizontal, Compound, Condensing Crank and Fly-wheel Pumping Engine at the Springfield, Ohio, Water Works. The engine is of the following dimensions:

Two H. P. steam cylinders	24	inches diameter.
Two L. P. steam cylinders	42	inches diameter.
Two pump-plungers (double-acting)	19½	inches diameter.
Two pump-plunger rods (single)	4	inches diameter.
All strokes	36	inches.]

To the Board of Trustees Springfield Water Works, Springfield, Ohio:

Gentlemen—I herewith submit to you the results obtained from data taken at the duty test of your new Gaskill Pumping Engine.

The trial commenced on August 30th, at 10.45 A. M., and continued for twenty-four hours, ending August 31st, at 10.45 A. M.

AVERAGES OF OBSERVED DATA.

Average steam pressure	75.35	pounds.
Average vacuum	27.1	inches.
Average water pressure by gauge	101.07	pounds.
Average water pressure by gauge	235.02	feet.
Average height of water in well below gauge	27.48	feet.
Total head pumped against	262.5	feet.
Capacity of pumps per revolution	182.25	gallons.
Total revolutions in 24 hours	28003.	
Average piston speed per minute	116.68	feet.
Total water pumped	**5,103,547**	**gallons.**
Total water pumped	42,529,558	pounds.
Total water fed in boilers	98,452	pounds.

The required duty when pumping at a capacity of **5,000,000** gallons as per contract, is **110,000,000 foot-pounds**, with a consumption of 100 pounds of coal, on the basis of ten pounds of water evaporated per pound of coal.

From the above data we have the duty thus:

$$\mathbf{113,395,447} = 1000 \times \frac{28003 \times 182.25 \times 8\tfrac{1}{3} \times 262.5}{98452}$$

In the above calculations no allowance is made for friction in the suction main. This was omitted, as it was not provided for in the contract.

There was no account taken of the amount of coal used, nor the temperature of feed-water, as the stipulated duty was based on 75 pounds steam pressure, and temperature of feed on a basis of evaporation as stated before.

For the dimensions of engine and pumps, would refer you to the specifications. [These dimensions are given above.]

It will be observed, in both duty and capacity performance, that the engines have fulfilled the contractor's guarantee.

Respectfully submitted,

CHARLES A. BAUER.

September 4, 1889.

I hereby certify that the above is a true copy.

WILLIAM R. SMITH, Secretary.

STATE OF OHIO, CLARK COUNTY, ss.

William R. Smith of Springfield, Clark County, Ohio, being first duly sworn, says he is Secretary of the Board of Trustees of the Springfield Water Works of Springfield, Clark County, Ohio, and that the above is a true copy of the report of Charles A. Bauer to said trustees, submitted by him to them on September 4, 1889.

WILLIAM R. SMITH.

Sworn to before me by William R. Smith, and subscribed by him in my presence, this September 6, 1889, as witness my hand and official seal on day and year above written.

A. P. LINN COCHRAN,

[Notary's Seal.] Notary Public, Clark County, Ohio.

SARATOGA SPR[illegible]

[illegible]

PREL[illegible]

PROF. DAVID M. GREENE [illegible]

INSTITUTE, ON THE S[illegible]

[illegible]

SAMUEL F. COREY, *Clerk Board of* [illegible]

DEAR SIR—A twenty-four hour [illegible] cluded at 12 noon yesterday.

I deemed it but just to the contractors [illegible] port, N. Y., and to your Water Board [illegible] handsomely performed all that was promised [illegible] of the engine is very considerably in excess [illegible] of the engine, based even upon the [illegible] excess of the guaranty.

The engine is an excellent piece of workmanship [illegible]

Of course I cannot at this time [illegible] during the first eighteen hours of the [illegible] 8,156,736 gallons in twenty-four hours [illegible] tion was about 111,000,000 [correction [illegible] assumed in the contract (10 [illegible] capable of maintaining a fire [illegible] conclusion, I beg to say that [illegible] the requirements of their contract [illegible] will submit a formal and complete [illegible]

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SARATOGA SPRINGS, N. Y.

OCTOBER, 1889.

REPORT OF PROF. DAVID M. GREENE.

To the Board of Water Commissioners of the Village of Saratoga Springs, N. Y.:

GENTLEMEN—I beg to report upon the recent test of the new Gaskill Horizontal Duplex Compound Pumping Engine, furnished to your village by the Holly Manufacturing Company of Lockport, N. Y., as follows:

DIMENSIONS OF ENGINE AND PUMPS.

The principal dimensions of the engine and pumps are as follows:

Diameter of high-pressure cylinders	27	inches.
Diameter of low-pressure cylinders	54	inches.
Diameter of pump plungers	25	inches.
Stroke of steam pistons and pump plungers	40	inches.
Diameter of high-pressure piston rods (steel)	3.5	inches.
Diameter of low-piston rods (2), (steel)	4.5	inches.
Diameter of pump rods	5	inches.
Diameter of crank shaft (fagoted iron)	12.5	inches.
Diameter of hub of crank	22.5	inches.
Depth of crank	11	inches.
Diameter of crank pins (steel)	7.5	inches.
Length of crank pins (steel)	9	inches.
Length of beam between centers	63	inches.
Length of upper beam pin	14	inches.
Diameter of upper beam pin	6	inches.
Length of lower beam pin	6	inches.
Diameter of lower beam pin	6	inches.
Diameter of fly-wheel	16	feet.
Depth of rim of fly-wheel	16	inches.
Width of face of fly-wheel	14	inches.
Weight of fly-wheel, about	28,000	pounds.

The steam cylinders are jacketed, both on the sides and ends. The jackets are supplied with live steam from the boilers, which, with the jacket condensation, passes on to the feed water heater, and there serves to heat the greater part of the feed water, which is taken from the main. The condensation in the jackets and heater is trapped off into the feed tank, whence the feed pumps, four in number, and single-acting, draw their supply.

The exhaust steam from the low-pressure cylinders passes to a Bulkley condenser, from which the warm water is wasted. Thus all of the heat contained in the hot-well, or its equivalent, is lost.

The clearance space in all of the cylinders is small, and is taken at 2.7 per cent. and 3 per cent. in the high and low-pressure cylinders, respectively.

The steam valves of the high-pressure cylinders are of the double-beat type, and the cut-off, which is practically instantaneous, is varied by the load on the pumps by means of the Holly automatic regulator.

The steam is admitted to, and exhausted from, the low-pressure cylinders through gridiron slide-valves.

The pumps, which are double-acting, are each fitted with 700 "Troy" valves, each of about 1½ inch diameter and ⅜ inch lift.

At each end of each pump, therefore, there are 175 induction and 175 eduction valves, giving an aggregate valve opening for the reception and discharge of the water equal to more than 0.6 of the effective area of the plunger.

This large valve area insures the passage of the water to and from the pumps with the minimum of frictional resistance. The loss of head, due to the passage of the water through the pumps ought not to be, and probably is not, greater than 0.25 of a foot.

An important advantage, resulting from the liberal valve opening, is the practicability of largely increasing the supply of water by increasing the speed of the engine, without the development of material frictional resistance in the valve openings.

BOILERS AND CHIMNEY.

Steam is furnished to this engine by two horizontal, cylindrical tubular boilers, of the following dimensions:

Number of boilers	2	
Diameter of shell	66	inches.
Length of shell	18	feet.
Thickness of shell	0.375	inch.
Thickness of heads	0.5	inch.
Domes, 2′ 9″ diameter by 3′ 8″ high.		
Number of tubes	87	
Diameter of tubes	3	inches.
Length of tubes	18	feet.
Width of fire grate	6	feet.
Length of fire grate	5.50	feet.
Area of fire grate (each boiler)	33	square feet
Height of chimney, above grate	98.5	feet.
Area of chimney, flue	8.33	square feet.

As originally set, the products of combustion passed from the furnaces back, under the boilers; forward, through the tubes, and back, over the tops, to the chimney connections at the rear ends. This has since been changed so that the products of combustion now pass directly from the tubes to a flue leading to the chimney, and thus do not pass over the tops of the boilers. This change has not resulted in an increased rate of evaporation; on the contrary, the rate of evaporation seems to have been somewhat diminished by the change.

From the foregoing dimensions, the following deductions are made as to the proportions:

Total area of grate surface	66	square feet.
Total heating surface (about)	2,866.0	square feet.
Total area of cross-section of tubes	7.18	square feet.
Total area of chimney flue	8.33	square feet.

Height of chimney above grate	98.5	feet.
Ratio of heating surface to grate surface	43.42	
Ratio of grate surface to area through tubes	9.19	
Ratio of grate surface to area of chimney flue	7.92	
Ratio of grate surface to area of flue of an equivalent chimney 60 feet high	6.2	

These proportions indicate that, with the moderate rate of combustion required for the supply of steam to your new engine, when running at its normal, or at the contract speed, there will be excessive draught; that the products of combustion must pass to the chimney at too high a temperature; or, that the dampers must be constantly and carefully regulated.

THE TEST.

The provisions of your contract with the Holly Manufacturing Company, as to capacity and duty, were:

1. That the capacity of the engine should be **8,000,000 U. S. gallons of** water, pumped against a pressure of 100 pounds per square inch, in 24 hours, at a piston speed of, or not exceeding, 115 feet per minute—equivalent to 17.25 revolutions per minute.

2. That, while in operation at the specified speed, and against the specified pressure, the engine should develop a duty of **105,000,000 foot pounds** per hundred pounds of coal consumed—the coal consumption to be based upon a hypothetical rate of evaporation of 10 pounds of water per pound of coal.

3. That the engine and pumps shall be capable of operating against a pressure of 140 pounds per square inch, with safety to all its parts.

The preliminary arrangements for the test were made by Mr. D. A. Decrow, representing the contractors, who also represented the contractors during the test.

It was found to be impracticable to *weigh* the feed water which was supplied to the boilers. It might, possibly, have been passed through a meter, and thus measured; but it was thought best to attach the only available meter to the pipe which took cold water from the main. Thus only the cold feed water was measured. The water, resulting from condensation in the jackets and heater, which was trapped off and joined the feed between the heater and the feed pumps, was not measured. It has been determined, however, with a fair degree of approximation, by estimating the condensation in the jackets and in the heater.

The coal, which had been hand-picked, to clear it from slate, was served in charges of 900 pounds each. The coal car, which weighed 360 pounds, was run upon the scales between charges to see that the scale beam remained properly balanced, and every necessary precaution was taken to insure fair and just conclusions from the test.

Indicator diagrams were taken every half hour from the high and low pressure cylinders and pump, of the left engine, from 12 M. to 6 P. M.; from 6 P. M. to 6 A. M. from the right engine, and from 6 A. M. to 12 M. of the 16th from the left engine.

The following data were observed every 15 minutes:

FIRE ROOM.

1. Steam pressure at the boilers.
2. Temperature of the feed water.
3. Temperature of the fire room.

Times of coal service, and amount served, and the quantity of feed water wasted, were also noted in the fire room.

ENGINE ROOM.

4. Steam pressure at the engine.
5. Water pressure at the engine.
6. Vacuum per engine gauge.
7. Reading of the water meter.
8. Reading of the engine counter.
9. Vacuum per gauge on condenser.

In making these records we were assisted by six young gentlemen of the graduating class of the Rensselaer Polytechnic Institute, who were divided into two watches of three persons each, who were relieved every four hours. The duty assigned to these gentlemen was faithfully and satisfactorily performed.

At 12 o'clock, noon, of the 15th, the machinery having been in operation several hours, proper normal conditions as to steam and water pressure, revolutions of engine, condition and depth of fires and height of water in the boilers having been established, the test was considered as begun.

The height of the water in the glasses was marked by cords tied to them, and the heights of these cords, above the bottoms of the glasses, were measured. A charge of 900 pounds of coal was served, and the fires were replenished.

It was intended that the duration of the test should be 24 hours, continuing until 12 M. on October 16th. But at about 7.30 A. M., of the 16th, it was discovered that the heater had burst, and that the steam and hot water were escaping in considerable quantities. The record showed that in all probability the fracture started sometime between six and seven A. M., and had been gradually enlarging. This accident caused us to reduce the duration of the test to 18 hours, and to regard it as having been concluded at six A. M., October 16th.

The conclusion of the test period at 6 A. M., was permissible, as well as necessary—because of the very uniform action of the machinery, and of the uniform rate of coal consumption, evaporation and height of water in the boilers. The twelfth charge of coal had been expended, and the fires were replenished from the thirteenth charge, at the beginning of the nineteenth hour, or at 6 A. M.

The engine was, nevertheless, kept in operation for the entire 24 hours, and the records were continued until noon of the 16th ; excepting, only, the readings of the water meter, which was in operation only during the last hour.

When the accident to the heater was discovered, it, and the meter, was thrown out of the circuit. Steam was also shut off from the jackets, and the boilers were supplied wholly with water from the main at 51° Fahr. This condition of things was established before 8 A. M., and it was hoped that a run of four hours might be had, with the necessary data, which would show the economic value of the jackets and heater. At 11 A. M., however, when the meter was finally put in operation, the water in the boilers was high—say $\frac{3}{4}$ inch ; while at 12 M., the water was down at its proper height. Thus the evaporation became too uncertain to justify an attempt at exact comparison. Nothing was certain, except that there was a marked increase in the rate of coal consumption during the last four hours of the run.

In Table No. 1 will be found the complete record of the first 18 hours' run, except as to the coal consumption.

Table No. 2 contains the record of coal consumption, with deductions therefrom.

From these tables, we extract the following values :

Mean steam pressure in boilers, per gauge	81.05	pounds
Mean steam pressure at engine, per gauge	78.01	pounds
Mean steam pressure in jackets, per gauge	70.075	pounds
Mean water pressure, per gauge	99.565	pounds
Mean vacuum, per gauge on condenser	28.9	inches
Mean vacuum, per gauge on engine	27.87	inches
Mean temperature of feed water, Fahr	203.55°	
Mean temperature of fire room, Fahr	62.57°	
Mean volume of water, at 51°, passing the meter per hour	88.014	cu. feet
Mean revolutions of engine, per minute	17.04	
Mean period of consumption of 900 pounds coal	1 h. 30 m.	
Mean rate of coal consumption, per hour	600.00	pounds
Mean rate of coal consumption, per minute	10.00	pounds
Mean rate of consumption, per hour, per square foot of grate	9.091	pounds

In order, now, to determine the proper values to be used in the estimate of the duty according to the terms of the contract, it is necessary to modify, or correct, the following values :

1. The mean water pressure, as indicated by the gauge.
2. The mean volume of water passing through the water meter.

1. *The mean water pressure.* The gauge on the engine was five feet above the pistons of the indicators on the pumps; representing 2.17 pounds.

The indicator diagrams taken from the pumps show that there was a lift, to the level of the indicator, of two pounds per square inch. We thus have :

Mean pressure, by the gauge	99.565	pounds
Pressure, due to five feet between indicator and gauge	2.17	pounds
Pressure below atmosphere on indicator	2	pounds
Total mean pressure on pumps	103.735	pounds

2. *Volume of water passing through meter, per minute.* A careful test of the water meter, made by Mr. Decrow on October 14th, showed that its indications were 4.17 per cent. too great. The temperature of the metered water was 51° Fahr.; at which temperature the weight of a cubic foot of water is 62.405 pounds.

The water wasted, in taking the temperature of the feed, was 567.25 pounds in 18 hours; or the waste was at the rate of 0.525 pounds per minute.

Then, the meter indication having been 88.014 cubic feet per *hour*, we have, for the weight of metered water per *minute :*

$$62.405 \times \frac{88.014}{60 \times 1.0417} = 87.873$$

This represents merely the water which passed the meter; to it should be added the water of condensation which subsequently entered the feed from the steam-jackets of the cylinders. It was not practicable to measure this last mentioned water, but from the records of a test of a similar engine, under like circumstances in 1887, at Philadelphia, where the jacket-water was measured, we find that this water amounted to almost exactly 10 per cent. of the metered water.

We see no reason why the relative amount of jacket-water should be different in this case; and, as the amount measured in Philadelphia is entirely reasonable, we assume that at Saratoga, in our test, the jacket-water was equivalent to 10 per cent. of the metered water. We thus obtain, finally:

Weight of metered water, per minute	87.873 pounds.
Add 10 per cent. for jacket-water	8.787 pounds.
	96.660 pounds.
Deduct waste in taking temperature	.525 pounds.
Mean rate of evaporation, per minute	96.135 pounds.

It should be stated, in this connection, that the *actual* evaporation of the boilers was greater than is indicated above; but the steam, resulting from this excess of evaporation, was condensed in the heater in heating the feed-water; and the heat which was brought from the boilers was returned to them in the feed-water. Thus, the performance of the engine and boilers was the same as it would have been, had there been no heater, and had all of the steam been supplied to the cylinders and their jackets—the jacket-water being added to the metered water and, with it, constituting the entire feed. In this latter case, the temperature of the feed would have been about 75° or 76° instead of 203 55°. This water would then have received all of its additional heat *in the* boilers instead of receiving part of it there and part of it in the *heater* at the expense of the *boiler*. In either case, the coal consumption would be the same.

It may well be asked: What useful purpose is served by the heater, in this case?

ESTIMATE OF DUTY.

For this purpose it will be convenient to construct a simple duty formula, as follows:

Area of pump plunger, 25 inches diameter	490.875 square inches.
Area of section of 5-inch rod = 19.635.	
Deduct one-half section of rod	9.8175 square inches.
Mean effective area of plunger	481.0575 square inches.

Under a presure of *one* pound per square inch, and at *one* revolution per minute, the net *work* of *two* pumps will be

$$481.0575 \times 2 \times \frac{2 \times 40}{12} = \ldots\ldots\ldots \quad 6414.1 \text{ foot-pounds.}$$

Then, if in any case,

r = number of revolutions per minute,

and p = mean pressure, in pounds per square inch on the pump plungers,

the foot-pounds, or work per minute, will be

$$6414.1 \text{ p. r.} \ldots\ldots\ldots \qquad \text{(a)}$$

Now, the *duty* is the number of foot-pounds due to the combustion of 100 pounds of coal, which, in the present case, is assumed to evaporate 10 pounds of water per pound, or 1,000 pounds of water per 100 pounds. The expression (a) must, therefore, be divided by the ratio of the *actual* rate of evaporation per minute to 1,000 pounds. Then D being the duty and W the number of pounds of water evaporated per minute, we have:

$$D = \frac{6414.1 \text{ p. r.}}{\frac{W}{1000}} = 6,414,100 \times \frac{\text{p. r.}}{W} \qquad \text{(A)}$$

Substituting the values already found in (A) we get, finally, for the duty:

$$D = 6,414,100 \times \frac{103.735 \times 17.04}{96.135} = \mathbf{117,936,698 \text{ foot-pounds.}}$$

This result exceeds the guaranteed duty **105,000,000** by **12.32 per cent.**, or nearly one-eighth.

The duty, based upon the **actual coal consumption**, may be found as follows:

C being the coal consumption per minute, in hundreds of pounds, the duty form will be:

$$D = 6414.1 \frac{p.\,r.}{c.} \quad \dots\dots\dots\dots\dots\dots \quad (B)$$

In the present case, the rate of coal consumption, per minute, was 10 pounds.

Whence $c = \frac{10}{100} = 0.1.$

This, in (B), gives us for the duty:

$$\begin{aligned} D &= 64141\ p.\,r. \\ &= 64141 \times 103.735 \times 17.04, \\ &= \mathbf{113{,}378{,}479\ foot\text{-}pounds.} \end{aligned}$$

This result is **7.98 per cent.** greater than the duty guaranteed by the Holly Ma facturing Company in their contract with your Board.

THE CAPACITY.

The capacity of the pumps of the new Saratoga engine is:

$$\frac{4 \times 481.0575 \times 40}{231} = 333.2 \text{ U. S. gallons per revolution.}$$

The rate at which water was pumped during the period of eighteen hours, was, therefore,

$$17.04 \times 333.2 \times 1440 = \mathbf{8{,}175{,}928\ gallons}$$

in twenty-four hours, at a piston speed of 113.6 feet per minute.

This rate, it will be observed, is something more than two per cent. greater than is required by the contract; while the pumps were operated against a pressure 3.73 per cent. greater than was required.

The quantity of water actually pumped during twenty-four hours, against a pressure of 103.575 pounds, and at a piston speed of 115 feet per minute, was **8,277,354 U. S. gallons**; exceeding the contract capacity by 3.47 per cent. against a pressure 3.57 per cent. greater than was required by the terms of the contract.

There is no doubt whatever as to the ability of this engine to operate the pumps against a pressure of 140 pounds per square inch, without danger to any of the parts of either engine or pumps. The pumps were not put under this pressure, for the reason that it was not deemed wise to endanger street mains, house plumbing or the hose, which discharged the surplus water into sewers, when no emergency existed which required or would justify the employment of such a pressure.

In the absence of a precise determination of the steam used in the cylinders and in their jackets, it has not appeared to be important that the large number of indicator cards taken should be thoroughly analyzed; but they have all been examined carefully, and the following facts have been deduced from the steam cards:

Mean distance which the steam followed in right high-pressure cylinder	0.294
Mean distance which the steam followed in left high-pressure cylinder	0.306
Mean distance which the steam followed in both high-pressure cylinders	0.300
Clearance of high-pressure cylinders equivalent to fraction of stroke	0.027
Mean volume of steam at cut-off, right high-pressure cylinder, fraction of piston displacement	0.321

Mean volume of steam at cut-off, left high-pressure cylinder, fraction of piston displacement	0.333	
Mean volume of steam at cut-off, both cylinders, fraction of piston displacement	0.327	
Volume of cylinders, plus clearance	1.027	
Steam expanded in right high-pressure cylinder, times	3.200	
Steam expanded in left high-pressure cylinder, times	3.085	
Steam expanded in both cylinders, times, mean	3.1425	
Mean pressure of steam, above zero, at cut-off in right high-pressure cylinder	95.952	pounds.
Mean pressure of steam, above zero, at cut-off in left high-pressure cylinder	95.44	pounds.
Mean pressure of steam, above zero, at cut-off in both cylinders	95.696	pounds.
Mean pressure above zero, end of stroke, right high-pressure cylinder	32.228	pounds.
Mean pressure above zero, end of stroke, left high-pressure cylinder	33.03	pounds.
Mean pressure above zero, end of stroke, both cylinders	32.629	pounds.
Clearance, low-pressure cylinders, fraction of stroke	0.030	pounds.
Mean pressure, at end of stroke, right low-pressure cylinder	7.86	pounds.
Mean pressure, at end of stroke, left low-pressure cylinder	7.653	pounds.
Mean pressure, at end of stroke, both cylinders	7.7565	pounds.

$$\frac{\text{Mean absolute pressure at cut-off} = 95.696}{\text{Mean absolute pressure end stroke} = 32.629} = 2.933.$$

$$\frac{\text{Mean absolute pressure at end of stroke, high-pressure cylinder} = 32.629}{\text{Mean absolute pressure at end of stroke, low-pressure cylinder} = 7.7565} = 4.207.$$

$$4.207 \times 2.933 = 12.349, \text{ by pressures.}$$

VOLUMES OF STEAM.

Volume of high-pressure cylinder, less half the rod	13.1423	cubic feet.
Volume of clearance, 2.7 per cent	.3548	cubic feet.
Volume of cylinder and clearance	13.4971	cubic feet.
Mean volume of steam at cut-off, including clearance, $= 0.3 \times 13.1423 \times 0.3548 =$	4.2975	cubic feet.
Mean expansion in high-pressure cylinders, by volume $= \frac{13.4971}{4.2975} =$	3.141	
Volume of low-pressure cylinder, less one rod	52.6463	cubic feet.
Volume of clearance, 3 per cent	1.5794	cubic feet.
Volume of clearance, high-pressure cylinder	.3548	cubic feet.
Final volume of steam	54.5805	cubic feet.
$\frac{\text{Final volume of steam} = 54.5805}{\text{Volume at end of stroke, high-pressure cylinder} = 13.4971} =$	4.044	
Total expansion, by volumes $= 4.044 \times 3.141 =$	12.702	times.
Mean absolute pressure, end stroke, high-pressure cylinder	32.629	pounds.
$\frac{\text{Mean absolute pressure at cut-off, high-pressure cylinder} = 95.696}{\text{Expansion in high-pressure cylinders} = 3.141} =$	30.467	pounds.
Mean absolute pressure, end stroke, low-pressure cylinder	7.7567	pounds.
$\frac{\text{Mean absolute pressure, at cut-off, high-pressure cylinder} = 95.696}{\text{Total measure of expansion} = 12.702} =$	7.534	pounds.

The mean pressure at cut-off in the high-pressure cylinder, as indicated by the gauge, is probably about three pounds in excess of the truth ; nevertheless, the foregoing deductions are of interest, as are the following approximate estimates of the water accounted for in the cylinders.

WATER ACCOUNTED FOR IN THE CYLINDERS.

. At the point of cut-off, in the high-pressure cylinders per minute :

$$4 \times 4.2975 \times 0.2212 \times 17.04 = 64.793 \text{ pounds.}$$

2. At end of stroke in the high pressure cylinders, per minute :

$$4 \times 13.4971 \times 0.0803 \times 17.04 = 68.474 \text{ pounds.}$$

3. At end of stroke in low-pressure cylinders, per minute:

$$4 \times 54.5805 \times 0.0207 \times 17.04 = 77.808 \text{ pounds.}$$

Assuming now, that the metered water, less the waste in taking the temperature of the feed, represents the water which entered the cylinders as steam, we have:

Pounds of water, per minute, passed the meter	87.873
Deduct waste	.525
Pounds of steam entered cylinders, per minute	87.348

Of this, there is accounted for at cut-off :

$$64.793 \text{ pounds or } \frac{64.793}{87.348} \times 100 = 73.95 \text{ per cent.}$$

At the end of stroke in the high-pressure cylinders:

$$68.847 \text{ pounds, or } \frac{68.847}{87.348} \times 100 = 78.82 \text{ per cent.}$$

And at the end of the stroke in the low-pressure cylinders:

$$77.808 \text{ pounds, or } \frac{77.808}{87.348} \times 100 = 89.08 \text{ per cent.}$$

Thus it appears that at the cut-off there was, in the high-pressure cylinders each minute, 22.555 pounds of water, constituting 26.05 per cent. of the steam and water which entered the cylinders. At the end of the stroke in the high-pressure cylinders, there appears to have been, each minute, 18.874 pounds of water, constituting 21.18 per cent. of the water and steam originally entering the cylinders.

At the end of the stroke in the low pressure cylinders, there was, each minute, 9.54 pounds of water, constituting 10.92 per cent. of the steam and water which originally entered the cylinders.

It is much to be regretted that such complete arrangements as would have enabled us to collect complete and precise data, could not be made.

With such data, it would have been practicable to have determined :

1. The water, if any, carried over with the steam—or the *quality* of the steam.
2. The condensation in the cylinders before cut-off, both with and without steam in the jackets.
3. The re-evaporation during expansion, in the high-pressure cylinders, under like conditions.
4. The re-evaporation in the low-pressure cylinders—under like conditions ; and
5. The quantity of water in the cylinders at the instant of exhaust.
6. The efficiency of the pumps, or the ratio of the net work of the pumps to the indicated work of the steam in the cylinders.

These facts, satisfactorily determined for this excellent type of engine, would be of great professional as well as public interest.

The materials used in the construction of the engine and pumps are apparently of the best; the workmanship is excellent, and the operation of the machine is, in all respects, simply admirable.

Our conclusion, therefore, is that the Holly Manufacturing Company have, in the performance of their contract with the Board of Water Commissioners of Saratoga Springs, materially exceeded their guaranty; and hence, that the machinery should be accepted.

In conclusion, while congratulating the citizens of Saratoga Springs upon the possession of what we think can be safely pronounced to be the best and most efficient pumping plant in the country, we desire to thank the Board of Water Commissioners, and their efficient and gentlemanly Clerk, Mr. Samuel F. Corey, for the generous provision which was made by them for our comfort during the test.

Respectfully submitted.

D. M. GREENE.

TROY. November 1. 1889.

TABLE No. 1.

RECORD OF TEST OF THE GASKILL HORIZONTAL DUPLEX COMPOUND PUMPING ENGINE AT SARATOGA SPRINGS, N. Y., OCTOBER 15 AND 16, 1889. CONTRACT CAPACITY, 8,000,000 U. S. GALLONS IN TWENTY-FOUR HOURS.

October 15.	PRESSURE OF STEAM, LBS.			Water Pressure, lbs.	VACUUM.		TEMPERATURES.		Reading of Water Meter, cubic feet.	Cubic feet of Water per hour at 51° F.	COUNTER.	
Hour.	In Boilers.	At Engine.	In Jackets.		Gauge on Cond.	Gauge on Engine.	Feed.	Fire rm.			Readings.	Revolu'ns per min.
12.00 M.	81	79.5	73	96	28.75	28.25	187°	74°	2416.5	....	903479	.....
12.15 P. M.	82	79	75	95	29	28.25	210.5	70.5	2424.0	....	903733	16.93
12.30 "	82	79	72	96.5	28.75	28.25	207	72	2447.25	....	903984	16.73
12.45 "	82	79	71	97	28.75	28.25	205	73	2471	....	904232	16.53
1.00 "	80	75	75	99.5	28.75	28	202	68	2494.5	78	904483	16.73
1.15 "	80	76	75	101	28.75	28	202	69	2519	....	904741	17.20
1.30 "	80	76	75	101	28.75	28	205	70	2541	....	904999	17.20
1.45 "	80	76	76	100.5	28.75	28	207	69	2562.75	....	905254	17
2.00 "	79	76	75	100	28.75	28	204	69.5	2585	90.5	905515	17.40
2.15 "	80	78	75	100	28.75	28	204	68	2608.5	....	905772	17.13
2.30 "	80	78	70	100.5	28.75	28	204	68	2629.5	....	906033	17.40
2.45 "	79	77	69	99.5	28.70	28	205	70	2650.5	....	906302	17.93
3.00 "	80	78	72	98.5	28.75	28	203	69	2671	86	906574	18.13
3.15 "	79	77	70	98.5	28.75	28	203.5	69	2694	....	906838	17.60
3.30 "	79	76	68	98.5	28.75	28	202	69	2717.5	....	907102	17.60
3.45 "	80	77.5	68	98.5	28.75	28	202	69	2741	....	907336	17.60
4.00 "	80	78	71.5	98	28.75	28	206	70.5	2763	92	907631	17.66
4.15 "	80	78	72	98	28.75	28	206	69	2784.5	....	907891	17.33
4.30 "	81	79	72.5	97	28.80	28	206	68.5	2805.25	....	908154	17.53
4.45 "	80	79	72	97	28.80	28	208	73	2828.5	....	908414	17.33
5.00 "	80	78.5	72.5	101	28.85	28	209	66	2848.50	85.5	908673	17.27
5.15 "	79	77	70	100	28.75	28	208	61	2870	....	908933	17.33
5.30 "	80	78	70	100.5	28.80	28	207	67	2892.5	....	909210	18.47
5.45 "	80	77	70	99.5	28.90	28.25	208	67	2912.875	....	909448	15.87
6.00 "	81	79	71	100.5	28.95	28.10	207	67	2933.625	85.125	909703	17.00

REMARKS.—The contract capacity is represented by 16.673 revolutions per minute.

RECORD OF TEST OF THE GASKILL HORIZONTAL DUPLEX COMPOUND PUMPING ENGINE AT SARATOGA SPRINGS, N. Y., OCTOBER 15 AND 16, 1889. CONTRACT CAPACITY, 8,000,000 U. S. GALLONS IN TWENTY-FOUR HOURS.

October 15.	PRESSURE OF STEAM, LBS.			Water Pressure, lbs.	VACUUM.		TEMPERATURES.		Reading of Water Meter, cubic feet.	Cubic feet of Water per hour at 51° F.	COUNTER.	
Hour.	In Boilers.	At Engine.	In Jackets.		Gauge on Cond.	Gauge on Engine.	Feed.	Fire rm.			Readings.	Revolu'ns per min.
6.15 P. M.	81.0	78	72	100	28.9	28.25	209°	60°	2954.6	......	909956	16.87
6.30 "	81	79	72	99	28.9	28.25	210	63	2975.625	......	910209	16.87
6.45 "	81	79.5	72.5	100.5	28.9	28.25	210	64	2996.875	......	910461	16.80
7.00 "	81	79	72	99.5	28.9	28.25	207	60	3018.25	84.625	910712	16.73
7.15 "	81	79	72	99.5	29	28.25	208	58	3039.25	......	910963	17.20
7.30 "	81	78	72	98.5	27	26.5	210	59	3059.95	......	911206	16.40
7.45 "	81	79	71.5	101	28.95	28	208	63	3081.125	......	911473	17.80
8.00 "	81	78	71	101	29.12	27.75	207	68	3102.875	84.625	911736	17.53
8.15 "	82	79	72	99.5	29	27.75	207	62	3124.125	......	912993	17.13
8.30 "	81	78.5	71	100.5	29	27.25	204	65	3146.125	......	912253	17.33
8.45 "	80	78	70	99.5	28.9	27.75	200	64	3170.125	......	912514	17.40
9.00 "	81	79	71	100.5	28.75	27.75	200	60	3194.125	91.25	912771	17.30
9.15 "	81	78.5	71	101	29	27.75	199	58	3218	......	913026	17
9.30 "	80	78	71	100	29	27.8	198	58	3241.5	......	913280	16.93
9.45 "	80	78	71.5	98.5	29	27.8	204	58	3265	......	913533	16.87
10.00 "	82	81	75	100.5	28.875	27.75	208	56	3281.75	87.625	913786	16.87
10.15 "	82	79	73	98.5	29	27.8	209	56	3297.125	......	914037	16.73
10.30 "	81	79	72	100.5	29	27.75	206	58	3318.5	......	914286	16.60
10.45 "	81	78	72	100.5	29	27.75	204	62	3341.25	......	914536	16.66
11.00 "	80	77	70	98	29	27.8	202	56	3364.125	82.375	914785	16.60
11.15 "	80	78	71.5	99.5	29	27.8	200	56	3387.625	......	915034	16.60
11.30 "	80	78	71.5	99	28.9	27.5	198	56	3411	......	915281	16.47
11.45 "	82	79.5	72	101	29	27.75	200	56	3434.25	......	915529	16.53
12.00 MID.	81	78	71	98.5	29	27.8	200	56	3457	92.875	915777	16.53
12.15 A. M.	80	77	70	98.5	29	27.8	197	60	3482	......	916025	16.53

TABLE No. 1.—*Concluded.*

RECORD OF TEST OF THE GASKILL HORIZONTAL DUPLEX COMPOUND PUMPING ENGINE AT SARATOGA SPRINGS, N. Y., OCTOBER 15 AND 16, 1889. CONTRACT CAPACITY, 8,000,000 U. S. GALLONS IN TWENTY-FOUR HOURS.

October 15.	PRESSURE OF STEAM, LBS.			Water Pressure, lbs.	VACUUM.		TEMPERATURES.		Reading of Water Meter, cubic feet.	Cubic feet of Water per hour at 51° F.	COUNTER.	
Hour.	In Boilers.	At Engine.	In Jackets.		Gauge on Cond.	Gauge on Engine.	Feed.	Fire rm.			Readings.	Revolu'ns per min.
12.30 A. M.	80	78	70	101	29	27.75	198°	58°	3507.25	......	916274	16.60
12.45 "	81	78	71	100.5	29	28.8	201	57	3528.75	......	916522	16.53
1.00 "	82	78.5	71	101	29	27.75	204	62	3550.5	93.50	916770	16.53
1.15 "	81	78	71	100.5	29	27.8	204	57	3572.75	......	917016	16.40
1.30 "	81	78	70.5	99.25	29	27.55	203	59	3595.50	......	917265	16.60
1.45 "	80	77	70	99	29	27.75	207	56	3617.50	......	917521	17.67
2.00 "	80	77	69	99.5	29	27.75	203	61	3639	88.50	917780	17.27
2.15 "	79	77	69	99.25	29	27.75	202	56	3661.25	......	918040	17.33
2.30 "	80	77.5	69	99.5	29	27.75	203	62	3683.50	......	918300	17.33
2.45 "	80	78	70	100.5	29	27.75	202	60	3706	......	918563	17.53
3.00 "	80	78	70	101	29	27.625	204	59	3728	89.00	918822	17.27
3.15 "	80	77.5	70	101	29	27.75	204	59	3750	......	919077	17
3.30 "	81	78	70.5	101.25	29	27.75	204	60	3772	......	919333	17.07
3.45 "	81	78	70.5	100.5	29.2	27.75	204	55	3794	......	919588	17
4.00 "	80	78	72	100	29	27.75	205	64	3816	88.00	919845	17.13
4.15 "	80	78	70	100	29	27.75	202	60	3837.75	......	920100	17
4.30 "	80	78	71	99	29	27.25	205	57	3869.5	......	920355	17
4.45 "	80	78	70	99.5	29.2	27.75	196	52	3883.5	......	920608	16.68
5.00 "	80	77	70	99.5	29.2	27.5	195	60	3909	93	920861	16.68
5.15 "	80	77	69.5	100.5	29.3	27.75	196	60	3934	......	921115	16.69
5.30 "	78	75	68	99.5	29.2	27.5	196	59	3958.25	......	921368	16.68
5.45 "	77	74	67.5	99.5	29.2	27.75	199	60	3980.5	......	921622	16.69
6.00 "	80	78	71.5	101	29.1	27.75	199	57	4000.75	91.75	921881	17.27
Means,	81.05	78.01	70.075	99.565	28.9	27.87	203°.55	62°.57	....	88.014	......	17.04

TABLE No. 2.

RECORD OF TEST OF THE GASKILL HORIZONTAL DUPLEX COMPOUND PUMPING ENGINE AT SARATOGA SPRINGS, N. Y., OCTOBER 15 AND 16, 1889.

COAL CONSUMPTION—WITH DEDUCTIONS.

DATE. 1889.	No. Charge, 900 lbs.	TIME OF CONSUMPTION.			RATE OF CONSUMPTION.		
		From	To	H. M.	Per hour, lbs.	Per min. lbs.	Per square foot grate per hour, lbs.
1	2	3	4	5	6	7	8
October 15..	1	12.00 M.	1.45 P. M.	1.45	514.3	8.57	7.793
" ..	2	1.45 P.M.	3.15 "	1.30	600	10.00	9.091
" ..	3	3.15 "	4.33 "	1.18	692.3	11.54	10.488
" ..	4	4.33 "	6.00 "	1.27	620.7	10.34	9.408
" ..	5	6.00 "	7.30 "	1.30	600	10.00	9.091
" ..	6	7.30 "	9.00 "	1.30	600	10.00	9.091
" ..	7	9.00 "	10.26 "	1.26	628	10.47	9.515
" ..	8	10.26 "	12.00 MID.	1.34	574.3	9.57	8.701
October 16..	9	12.00 MID.	1.35 A. M.	1.35	568.6	9.48	8.615
" ..	10	1.35 A. M.	3.00 "	1.25	635.1	10.58	9.623
" ..	11	3.00 "	4.25 "	1.25	635.1	10.58	9.623
" ..	12	4.25 "	6.00 "	1.35	568.6	9.48	8.615
Means......	..	....	....	1.30	600.00	10.00	9.091

NOTE.—The "Means," in the above table, are the means for the entire period of eighteen hours, and not of the rates in the several columns, which obtain for the different times indicated in the fifth column.

DAYTON, OHIO.

NOVEMBER, 1889.

REPORT OF ARTHUR GIESLER, M. E.

To the Trustees of the Water Works of the City of Dayton, Ohio:

GENTLEMEN—Herewith I have the honor of handing you the report upon the duty and performance of the lately erected Gaskill Compound Pumping Engine, bought of the Holly Manufacturing Company of Lockport, N. Y.

The engine is of the horizontal beam, rotative, compound, condensing type, coupled to a fly-wheel, shaft with cranks at ninety degrees. The high-pressure cylinder is mounted on top of the low-pressure cylinder. The low-pressure cylinder is firmly bolted to a bed frame, with the pistons moving in opposite directions. The pumps are in line with the low-pressure cylinder and bolted to the same bed frame. The steam cylinders are jacketed, including heads with steam of boiler pressure, and the condensation is trapped through a heater to the boiler feed pumps.

The contract requirements of the engine were a pumping capacity of **10,000,000 United States gallons** in twenty-four hours, working at nineteen revolutions per minute against a pressure of sixty pounds per square inch, including suction-lift, and to develop a duty of **105,000,000 pounds** of water raised one foot high, with a consumption of 100 pounds of coal, on the basis of ten pounds of water evaporated by a pound of coal.

The test began November 22, 1889, at 10.30 A. M., and terminated November 23, 1889, 10.30 A. M.—a period of twenty-four hours. The platform scales weighing the water to the boilers, were tested and sealed the previous date. The pressure gauge was tested shortly before the trial by a certified test gauge number.

Having no means of measuring the actual amount of water pumped without shutting off the city supply, computations were agreed upon a basis of plunger displacement, deducting three per cent. for slip and other defective pump actions, a deduction which the representative of the Holly Manufacturing Company thought entirely too great, but finally agreed to sanction it. The condensation water from the steam-jackets is, as stated before, carried back to the boilers. To ascertain this amount exactly would necessitate a large expense of piping, and, having sufficient data on hand of previous tests (see Philadelphia test, January 10, 1888), it was agreed to make it ten per cent. of the total water evaporated in the boilers during the twenty-four hours and charged to the engine.

It was agreed to take half hourly observations from boiler pressure gauge, water pressure gauge, vacuum gauge, revolution indicator, and readings of difference of level between center of pressure gauge and surface of supplying well. To facilitate these readings, said difference was found by leveling with a leveling instrument and a constant of nineteen feet, six and three-eighths inches thus established was used by plus and minus readings to arrive at the proper difference.

The water for the boilers was taken from the city supply and piped to a wooden tank mounted on a platform scale capable of holding one thousand pounds, weighed and run thence to another tank below of the same capacity, from which a pipe attachment was made to the two back pumps feeding the boilers. These pumps are driven from engine rocker shaft. Each water charge was 1,000 pounds, and as the specifications required that ten pounds of water should be considered one pound combustible, therefore particular attention was given to the accuracy of the observation on the amount of water weighed to the boilers.

Mr. Clarence Folsom was in charge of this part of the trial in behalf of the city. Relent to one pound of coal, made the engines and boilers two separate plants.

The boilers furnishing steam to the engine were three in number, of the horizontal tubular type, and used for a period of many years to supply steam for the old engines. The management of the boilers and all attachments to and from were under the able supervision of the Chief Engineer, Mr. Martin Weaver; and all the data given about evaporation capacity of said boilers are taken from the memoranda of that gentleman.

The management of the engines during the trial, was under the directions of the representative of the Holly Manufacturing Company, Mr. Decrow.

During the entire test the performance of the engine was uniformly satisfactory in every observed particular. From the readings of the pressure gauge you can observe the pressure to be nearly uniform with very small variations; the vacuum gauge was nominally constant, and the steam pressure gauge shows the same uniformity which is partly due to the excellent valve-gear, partly to the proper management of the boilers, and greatly to the accomplished skill in firing of your Mr. John King.

By some neglect in oiling, the upper pin in the left-hand rotative beam was heated, but in about five hours the pin was cooled down and continued to be for the duration of the trial. Stuffing boxes and glands were perfect; no escape of steam, neither from high or low-pressure cylinder could be noticed. The only noise that could be heard was made by the air pumps, on account of the brasses being a trifle loose.

The indicator cards taken jointly by Mr. Decrow and Mr. Lufkin from the high and low-pressure cylinders show an evidence of perfection of this sensitive valve-gear, and if there is any cause for criticism, then it may be mentioned that the induction corners on both high and low-pressure cards could be sharper. However, it must be mentioned also that the valve-gears adjusted for the daily working speed, for a speed two-thirds less than the testing speed, and I venture to say that cards taken at a speed of twelve to fifteen revolutions per minute would demonstrate the beauty of this valve-gear to its full extent.

Enough cannot be said about the regulator. Any variation in pressure, the regulator will correct in one to one and a half revolutions of the engine.

As to the expansion curves on indicator cards, I believe they are true to their theoretical expansion curve. The loss between high and low-pressure cylinders is very small; the amount of this, however, I cannot ascertain exactly for want of data to plot the cards to one scale. If you wish, however, I will procure the percentage of clearance, etc., and hand it to you at the earliest moment.

The indicator cards taken from pumps are perfect. The waved corners are the result of the vibration of the indicator spring. The waved line in middle of stroke is due to vibrations of the valves.

In the summary of the test readings you will find 840 pounds of water deducted from the total of evaporation. The amount of this derives from the collection of four leakages of the steam valve leading to the old engines that could not be made perfectly tight.

I will not omit to mention that the test was made in pumping against sixty pounds pressure per inch, *excluding* suction-lift, which means the pumps were working against an additional pressure of about twenty feet, not provided for in the contract.

In consideration of the remarkable duty of this engine, I think it necessary to append for future reference correct copies of all readings and observations, indicator cards from all pistons and plungers.

For information as to essential points of the test you are respectfully referred to the following, viz.: Description, summary, result, and evaporation.

DESCRIPTION.

Taken from specifications.

High-pressure cylinder	25	inches diameter.
Low-pressure cylinder	50	inches diameter.
Pump plungers	27½	inches diameter.
Stroke of all pistons and plungers	38	inches.
Plunger rods	4¾	inches diameter.
Main crank shaft	11¼	inches diameter.
Fly-wheel	16	feet diameter.

SUMMARY.

Duration of test	24	hours.
Average mean pressure by gauge	60.01	pounds.
Mean difference in level between center of pressure gauge and source of supply	19.46	feet.
Average total head pumped against	158.023	feet.
Revolution of engine during trial	27,047	
Plunger displacement per revolution of engine in United States gallons	385	gallons.
Feed water weighed to the boilers	96,818	pounds.
Jacket water returned to boilers, 10 per cent. of total	10,664.22	pounds.
Total water weighed to boilers in 24 hours	107,482.22	pounds.
Total leakage of water from leaks	840	pounds.
Total water (steam) used by engine in 24 hours	106,642.22	pounds.
Temperature of well water	52°	Fahr.
Temperature of feed water to boilers	53°	Fahr.
Average steam pressure	75.82	pounds.
Average vacuum	26.45	inches.
Contract basis of evaporation	10	pounds.
Weight of one United States gallon of water at 52° Fahrenheit	8.337	pounds.
Temperature of air in engine-room	70°	Fahr.

RESULT.

$$\text{Duty} = \frac{385 \times .97 \times 27.047 \times 8.337 \times 158,023 \times 1,000}{106,642.22} = 124,782,157 \text{ foot-pounds.}$$

Contract duty	105,000,000
Excess	19,782,157
or 18.8 per cent. above guarantee.	
Capacity, 385 × .97 × 27,047	10,100,702.15 gallons.
Capacity as per contract	10,000,000 gallons.
Excess	100,702.15 gallons.

EVAPORATION.

Feed water weighed into three boilers	96,818	pounds.
Total coal consumed	20,100	pounds.
Ashes and incombustibles weighed back	630	pounds.
Temperature of feed water	53°	Fahr.
Average steam pressure	75.82	pounds.

Evaporation per pound of coal, $\frac{96,818}{20,100-630} = 4.811$ pounds.

Evaporation from and at 212° Fahrenheit, $\frac{1211-53}{966} \times 4.811 = 5.763$ pounds.

The result of evaporation is not satisfactory. A fair average will be seven to eight pounds, but in order to pass a more correct judgment and to locate the deficiency, a more elaborate test would be necessary. The boiler test was a secondary matter. No calorimeter tests were made, and possibly the trial would show better results if only two instead of three boilers had been used.

Recapitulating the result, the engine has fulfilled all the requirements of the contract, and the City of Dayton should be congratulated upon the installation of so fine a pumping plant.

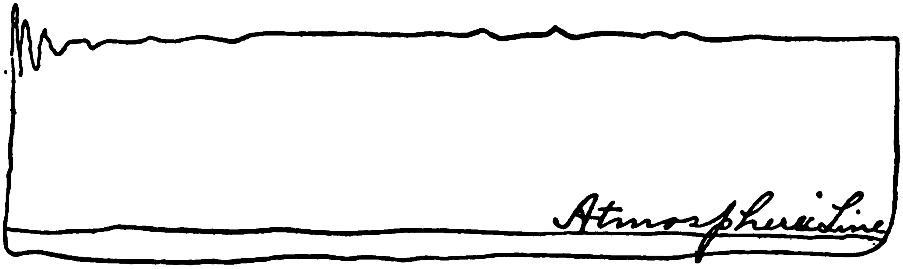
Respectfully submitted.

November 30, 1889. ARTHUR GIESLER, M. E.

Indicator Cards Taken During Trial Test November 22 and 23, 1889.

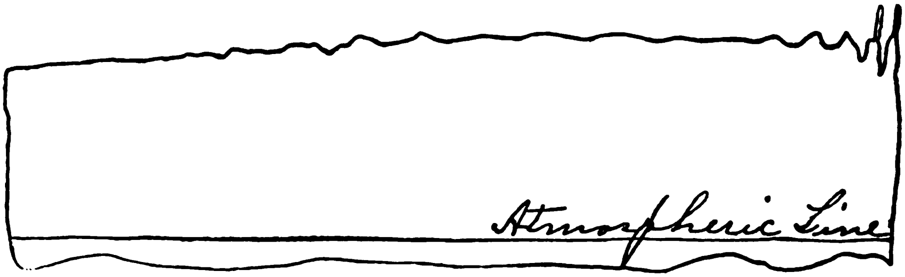
INDICATOR CARD ON PUMP TAKEN 8.45 A. M., NOVEMBER 23, 1889.

Back End. 60 *Spring.* 18½ *Revolutions.*



INDICATOR CARD ON PUMP TAKEN 8.45 A. M., NOVEMBER 23, 1889.

Front End. 60 *Spring.* 18½ *Revolutions.*

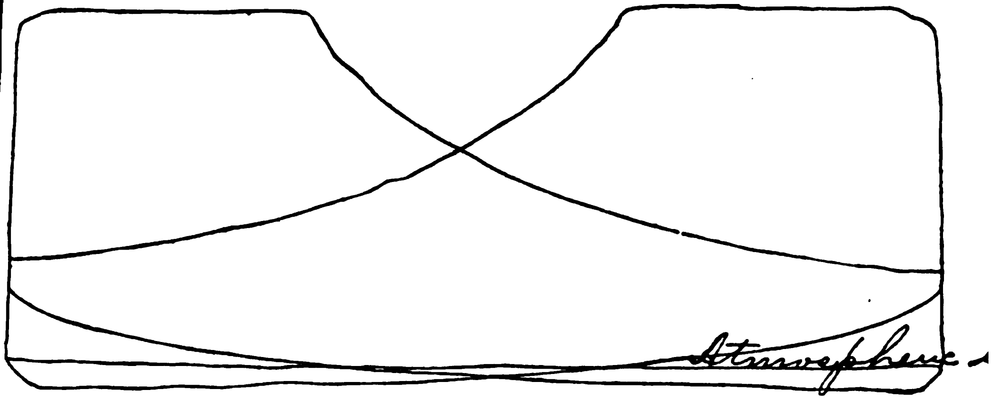


NOVEMBER 30, 1889. ARTHUR GIESLER, M. E.

Indicator Cards Taken During Trial Test November 22 and 23, 1889.

INDICATOR CARD TAKEN 8.45 A. M., NOVEMBER 23, 1889.

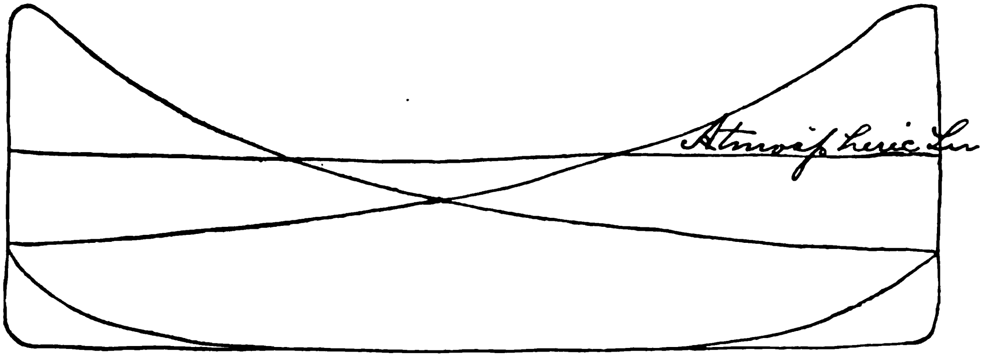
Back End. 40 *Spring.* *Front End.*



High Pressure Cylinder 18½ *Revolutions.*

INDICATOR CARD TAKEN 8.45 A. M., NOVEMBER 23, 1889.

Back End. 12 *Spring.* *Front End.*



Low Pressure Cylinder 18½ *Revolutions.*

NOVEMBER 30, 1889. ARTHUR GIESLER, M. E.

GAUGE READINGS DURING TRIAL TEST NOVEMBER 22 AND 23, 1889.

Time.	Pressure	Difference in level.	Steam Pressure	Vacuum.
	Pounds.	Feet.	Pounds.	Inches.
A. M.				
10.30	60	19.65	78	26¼
11.00	60	19.80	75	26¼
11.30	60	19.88	76	26¼
12.00	59½	19.99	75	26¼
12.30	60	19.95	76	26¼
P. M.				
1.00	60	19.90	75	26¼
1.30	59½	19.99	76	26¼
2.00	60	20.03	76	26¼
2.30	59	20.05	76	26¼
3.00	60	20.19	73	26¼
3.30	60	20.19	73	26¼
4.00	59½	20.28	76	26¼
4.30	60	20.34	75	26¼
5.00	60½	20.39	77	26¼
5.30	61	20.36	77	26¼
6.00	60	20.28	75	26¼
6.30	60	20.11	77	26½
7.00	60½	19.96	77	26½
7.30	60	19.90	74	26½
8.00	60	19.78	77	26½
8.30	60	19.71	77	26½
9.00	60	19.63	75	26½
9.30	60	19.53	75	26½
10.00	60	19.44	76	26½
10.30	60	19.40	75	26½
P. M.				
11.00	60	19.28	76	26½
11.30	60	19.24	76	26½
12.00	60½	19.20	77	26½
A. M.				
12.30	60	19.09	75	26½
1.00	60	19.08	76	26½
1.30	60	18.94	75	26½
2.00	60	18.84	76	26½
2.30	60	18.82	76	26½
3.00	60	18.76	75	26½
3.30	60	18.71	76	26½
4.00	60	18.67	75	26½
4.30	60	18.69	78	26½
5.00	60	18.69	76	26½
5.30	60	18.69	77	26½
6.00	60	18.69	75	26½
6.30	60	18.69	76	26½
7.00	60½	18.69	74	26½
7.30	60	18.69	77	26¾
8.00	60	18.69	78	26¾
8.30	60	19.07	76	26⅔
9.00	60	19.24	76	26¾
9.30	60	19.41	75	26¾
10.00	60	19.53	75	26¾
10.30	60	19.53	77	26¾
Mean Average	60.01	19.46	75.82	26.45

Pressure on gauge, 60.01 pounds × 2.309 = 138.563 ft.

Difference in level.................... 19.46

Total head pumped against...... 158.023 ft.

ARTHUR GIESLER, M. E.

NOVEMBER 30, 1889.

WATER CHARGES DURING TRIAL TEST, NOVEMBER 22 AND 23, 1889.

Water Charges.		Water Charges.		Water Charges.		Water Charges.	
Time.	Pounds.	Time.	Pounds.	Time.	Pounds.	Time.	Pounds.
A. M.		P. M.	25,000	A. M.	49,000	A. M.	73,000
10.30	1000	4.56	1000	12.08	1000	5.57	1000
10.47	1000	5.13	1000	12.30	1000	6.10	1000
11.08	1000	5.28	1000	12.53	1000	6.20	1000
11.22	1000	5.45	1000	1.10	1000	6.30	1000
11.42	1000	6.00	1000	1.30	1000	6.40	1000
11.54	1000	6.15	1000	1.40	1000	6.52	1000
P. M.		6.30	1000	1.55	1000	7.00	1000
12.10	1000	6.45	1000	2.05	1000	7.12	1000
12.26	1000	7.00	1000	2.20	1000	7.23	1000
12.40	1000	7.23	1000	2.35	1000	7.35	1000
12.55	1000	7.40	1000	2.56	1000	7.45	1000
1.07	1000	8.00	1000	3.18	1000	7.55	1000
1.22	1000	8.15	1000	3.40	1000	8.07	1000
1.40	1000	8.33	1000	3.55	1000	8.16	1000
1.52	1000	8.52	1000	4.10	1000	8.28	1000
2.10	1000	9.08	1000	4.20	1000	8.40	1000
2.23	1000	9.30	1000	4.40	1000	8.50	1000
2.40	1000	9.50	1000	4.46	1000	9.00	1000
2.55	1000	10.15	1000	4.52	1000	9.15	1000
3.10	1000	10.30	1000	5.03	1000	9.35	1000
3.25	1000	10.55	1000	5.15	1000	9.48	1000
3.45	1000	11.10	1000	5.25	1000	10.00	1000
4.00	1000	11.30	1000	5.35	1000	10.15	1000
4.15	1000	11.50	1000	5.46	1000	10.30	818
4.28	1000						
4.44	1000						
Totals..	25,000	....	49,000	....	73,000	....	96,818

Total water charge during test..........................96,818 pounds.

ARTHUR GIESLER, M E.

November 30, 1889.

BINGHAMTON, N. Y.

FROM THE ANNUAL REPORT OF THE WATER COMMISSIONERS FOR THE YEAR 1889.

[The following extracts from the aforesaid report are given as showing remarkably interesting results in daily service, with the Holly Quadruplex, Compound, Condensing Pumping Engine, as applied to the Holly system of water supply and fire protection for cities and villages, which was invented by Birdsill Holly and promoted by the Holly Manufacturing Company. The leading feature of the system is that of combining low water pressure for domestic supply with high water pressure for fire protection, which two objects or results are accomplished with one and the same plant, by the use of suitable machinery with closed mains and pipes. The system has been successfully introduced in more than two hundred cities and towns since 1878, and is still in use with, as the following extract should prove, satisfactory results:]

COST OF PUMPING SERVICE.

Salary of engineers	$1,932.00
Salary of firemen	1,200.00
Salary of assistant fireman	250.00
Fuel (including some fuel not used)	2,435.07
Gas	174.58
Oil and lubricators	184.51
Packing and gaskets	81.65
Compound for boilers	103.64
Waste	29.65
Telephone company	46.00
Repairs for boiler, engine, etc.	298.00
	$6,735.61

Pressure was raised for fires forty times during the year.

SUMMARY OF STATISTICS.

Population, by census of 1880	17,317	
Population (estimated)	35,000	
Kind of engines—two Holly quadruple, 4 cylinder, compound, condensing.		
Capacity—2,000,000 and 6,000,000 gallons per 24 hours.		
System—direct pressure.		
Source of water—excavated wells and Susquehanna River.		
Number of days pumping	365	
Average pumping time per day	24	hours.
Total number gallons pumped during year	1,093,747,691	
Total number gallons pumped per month	91,145,640	
Total number gallons pumped per day	2,996,569	
Number of gallons pumped per one cent fuel	5,385	
Average head pumped, in feet	166	
Dynamic head on pump by gauge	60	pounds.
Total amount of coal consumed during the year	1,123	tons.
Total amount of coal consumed per day	6,152	pounds.
Average pounds of ashes per day	No record.	
Kind of coal—Anthracite buckwheat	1,123	tons.
Cost of coal per ton	$ 1.85	
Cost of wood for the year	10.00	
Cost of fuel for the year	2,087.55	
Cost of fuel to raise 1,000,000 gallons 166 feet high	$1.98	
Cost of fuel to raise 1,000,000 gallons 1 foot	**$0.011**	
Duty in pounds of water raised one foot high with 100 lbs. coal	62,968,000	

WORK DONE BY THE ENGINES AND PUMPS FOR THE YEAR ENDING DECEMBER, 1889.

MONTHS. 1889.	Gallons Pumped Monthly.	Gallons Pumped Daily.	Height of Water in Wells at noon.		Domestic Pressure.	Coal Consumed Monthly.	Coal Consumed Daily.	Number of Fires.	Duration of Fires		Fire Pressure.	Gallons Pumped at Domestic Pressure.	Domestic Head plus one pound friction in suction.	Gallons Pumped at Fire Pressure.	Fire Head plus one friction in suction.	Duty.
	Gallons.	Gallons.	Ft.	In.	Lbs.	Lbs.	Lbs.		H.	M.	Lbs.	Gallons.	Feet.	Gallons.	Feet.	Gallons.
January	84,305,715	2,719,539	4	3	50	168,000	5,419	7	5	20	80	83,598,515	143.5	707,200	212.7	60,156,000
February ...	85,644,530	3,058,733	4	9	50	178,200	6,364	6	4	40	80	85,009,530	143.	635,000	212.2	57,400,000
March	87,086,750	2,809,250	4	9	50	165,200	5,329	6	6	20	80	86,243,550	143.	843,200	212.2	63,013,000
April	82,611,900	2,753,730	4	9	50	176,600	5,887	3	4		80	82,067,900	143.	544,000	212.2	56,000,000
May.... ...	85,189,374	2,748,044	5	3	50	168,500	5,485	3	1	25	80	84,996,774	142.5	192,600	211.4	60,000,000
June.........	90,580,726	3,019,357	3	9	60	200,500	6,683					90,580,726	167.			62,700,000
July.........	97,710,264	3,151,944	4	6	60	192,700	6,216	2		35	80	97,630,964	165.6	79,300	212.45	69,800,000
August	98,550,240	3,179,040	3	6	60	212,000	6,838	4	3	45	80	98,040,240	166.6	510,000	213.45	64,530,000
September...	98,259,600	3,275,320	4	9	60	196,500	6,550	2		50	80	98,146,270	166.	113,330	212.2	69,080,000
October.....	95,436,532	3,078,598	4	9	60	213,600	6,890	2	1	10	80	95,277,872	166.	158,660	212.2	61,737,000
November ..	94,314,960	3,143,832	4	8	60	176,250	5,875	1		30	80	94,246,960	166.1	68,000	212.3	65,180,000
December...	94,057,100	3,034,100	4	3	60	197,600	6,374	4	2	35	80	93,805,770	166.6	251,330	212.7	66,020,000
Total.......	1,093,747,691							40	31	10		1,089,645,071		4,102,620		
Average		2,996,560				2,245,650	6,155						156.5		213.1	62,968,000

Cost of raising 1,000,000 gallons one foot high, $3\frac{65}{100}$ cents. Cost of coal and wood $2,031.08.

Gallons pumped per pound of coal consumed, 487. Cost of pumping service [including coal and wood] $6,269.62.

BRANTFORD, ONT.

JANUARY, 1890.

REPORT OF THOMAS WORSWICK, M. E.

To the Chairman of the Water Commissioners of the City of Brantford:

GENTLEMEN—I have the honor to report to you on the performance of the pair of compound condensing Gaskill pumping engines, contracted for by the Waterous Engine Works Company of your city, and manufactured by the Holly Manufacturing Company, Lockport, N. Y.

The engines are of the well-known Gaskill design, with high-pressure cylinders of twelve inches diameter, low pressure cylinders twenty-four inches in diameter, and pump cylinder fourteen inches in diameter, twenty-two inch stroke.

The test was begun on the 8th of January, at ten A. M., and extended over a period of twenty-four hours. The method adopted was to weigh the water, which was pumped into a barrel placed on the platform scale, and run from this barrel into an iron tank, from which the water was pumped into the boiler by an independent feed-pump. As there was no means provided by which the water could be fed to the boiler by the pumps attached to the engine, I was compelled to use the independent pump for that purpose, which is very wasteful of steam. I also found it impossible to collect the water from the jacket of the cylinders, and, as I could not provide means for collecting it without a very considerable delay, it was therefore decided to go on with the test, although the conditions were very unfavorable for getting the best duty from the engine, as the result of my calculations will show.

Only one boiler was used, and before starting on the test the height of the water in the boiler was noticed and left at the same height at the close of the test. It was found, however, at the termination of the trial, that the water in the boiler next the one used had ¾ of an inch. This could only have come from a leak in the stop-valve connecting the two boilers together, and must be credited to the engine as well as the water used by the steam pump, and that lost from the jacket of the cylinders.

The dimensions of the engines and pumps, with the data for the calculations, are as follows:

High-pressure cylinder, diameter	12	inches.
Low-pressure cylinder, diameter	24	inches.
Pump cylinder, diameter	14	inches.
Pump piston-rod, diameter	2½	inches.
Mean area of pump cylinder	151.49	
Mean pressure of water gauge from 49 readings	61.795	pounds.
Mean distance from water in well to floor of pump well, 6.478 =	2.808	pounds.
Distance from center of pump cylinder to pressure gauge, 6.25 =	2.109	pounds.
Frictional resistance in suction-pipe and passages	1.168	pounds.
Frictional resistance between pump cylinder and pressure gauge	.422	pounds.
Total pressure on pump plunger per square inch	71.336	pounds.

Total pressure on both (combined) pump plungers	21.613.2	
Engine counter at beginning of trial	225	
Engine counter at termination of trial	35,890	
Total number of revolutions in twenty-fours hours	35,665	
Total number of feet traveled by both pump plungers	261,543.32	
Total water feed to boilers	34,646	pounds.
Water used by feed pump five per cent. of the whole	1,732	pounds.
Water lost from the jackets of cylinders, 10 per cent	3,464	pounds.
Water leaked through to the other boiler, 226 pounds per hour	5,428	
Total water used by the engine	29,218	pounds.
Gallons of water displaced per revolution	57.7	
Total revolutions for the trial of twenty-four hours	35,665	

The duty calculation becomes

$$151.49 \times 71.336 \times \frac{261543\ 32}{29218} \times 100 = \mathbf{96,735,491.16}$$

foot-pounds with 100 pounds of coal consumed, based upon a boiler performance of an evaporation at 10 pounds of water per pound of coal.

The capacity calculation is 57.7 × 35665 = **2,057,870.5 U. S. gallons.**

The specifications say, "The machinery above described will be guaranteed to be capable of lifting 2,000,000 U. S. gallons of water per twenty-four hours against a head of sixty pounds for domestic service, and against a head of 150 pounds for fire service at a piston speed not to exceed ninety feet per minute, and will be guaranteed to perform a test duty of 95,000,000 foot-pounds from each 100 pounds of coal consumed, based upon a boiler performance of an evaporation of ten pounds of water per pound of coal." This part of the contract has been fulfilled to the letter. The engine, pumps, and all the details pertaining thereto, are strictly first-class and complete in every respect.

At the termination of the twenty-four hour test, commenced to run the Gaskill engines on the fire test. The engines were started to work against sixty pounds pressure, and the pressure gradually increased until it rose to 150 pounds. I, of course, expected the engines to show some signs of labor as the pressure pumped against increased, but to my surprise they moved along with as much apparent ease as when pumping for domestic service with only sixty pounds pressure. This test only lasted two hours, for the reason that I became fully satisfied that the engines would perform this duty for any length of time without the slightest risk to any part of the machinery.

From the appearance of the indicator diagrams taken from these engines, I was led to expect a duty of at least 100,000,000 foot-pounds with 100 pounds of coal. The duty not coming up to what I looked for, I made a careful estimate of the water consumed by the engines, as accounted for by the indicator, and I find that had there been no leakage of water, and if the water lost by the jackets with its heat had been returned to the boiler, as will be the case when in constant use, with the saving of steam effected by using the pumps on the engines to feed the boiler, the duty would have been considerably over 100,000,000 foot-pounds.

The specification for boilers is no specification at all. It appears to have been left to the honor of the contractors as to the quality of the boilers supplied. I can only say that had the contractors been bound by the most stringent specifications possible they could not have given you better value.

All of which is respectfully submitted,

THOMAS WORSWICK.

HAMILTON, January 27, 1890.

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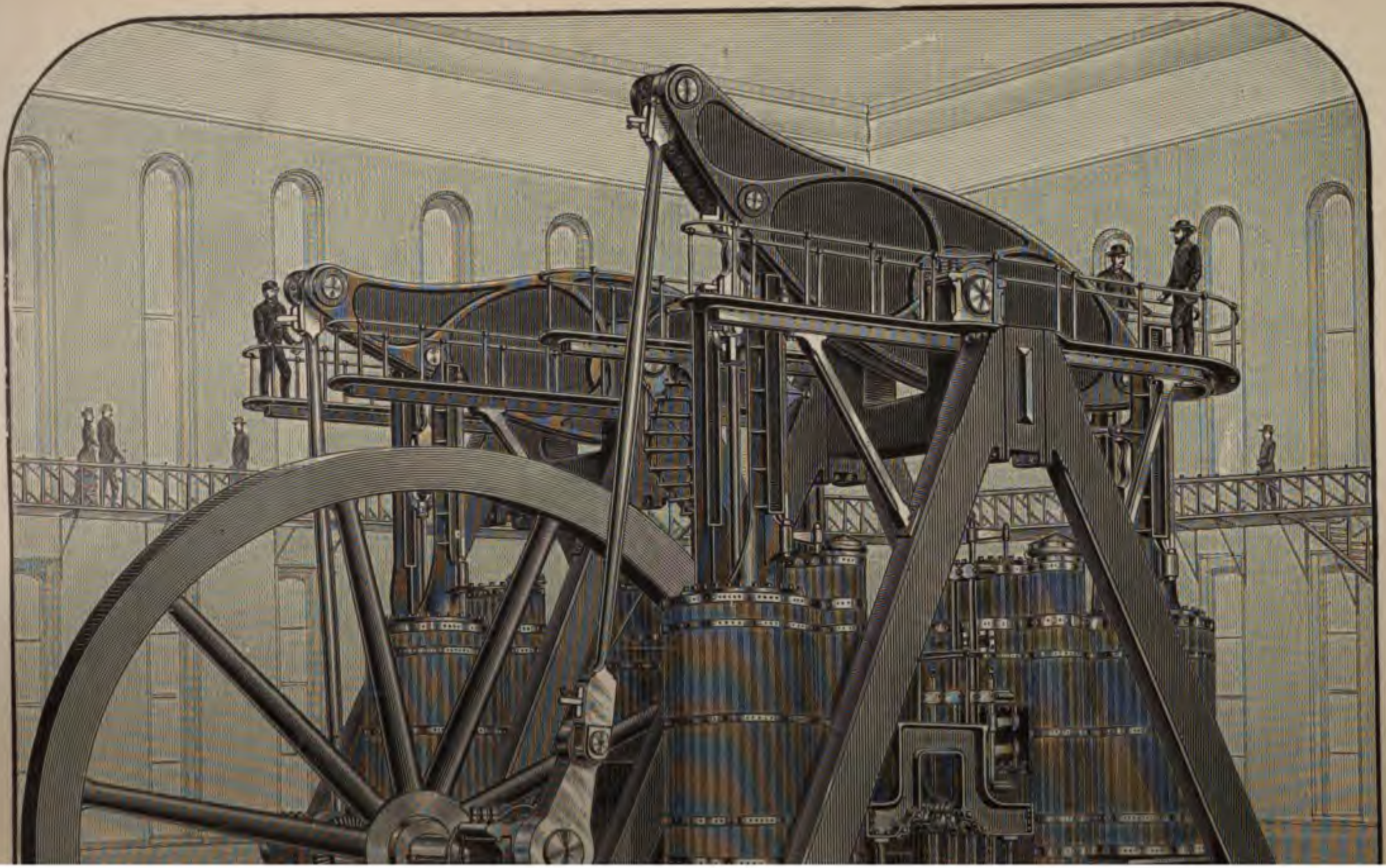

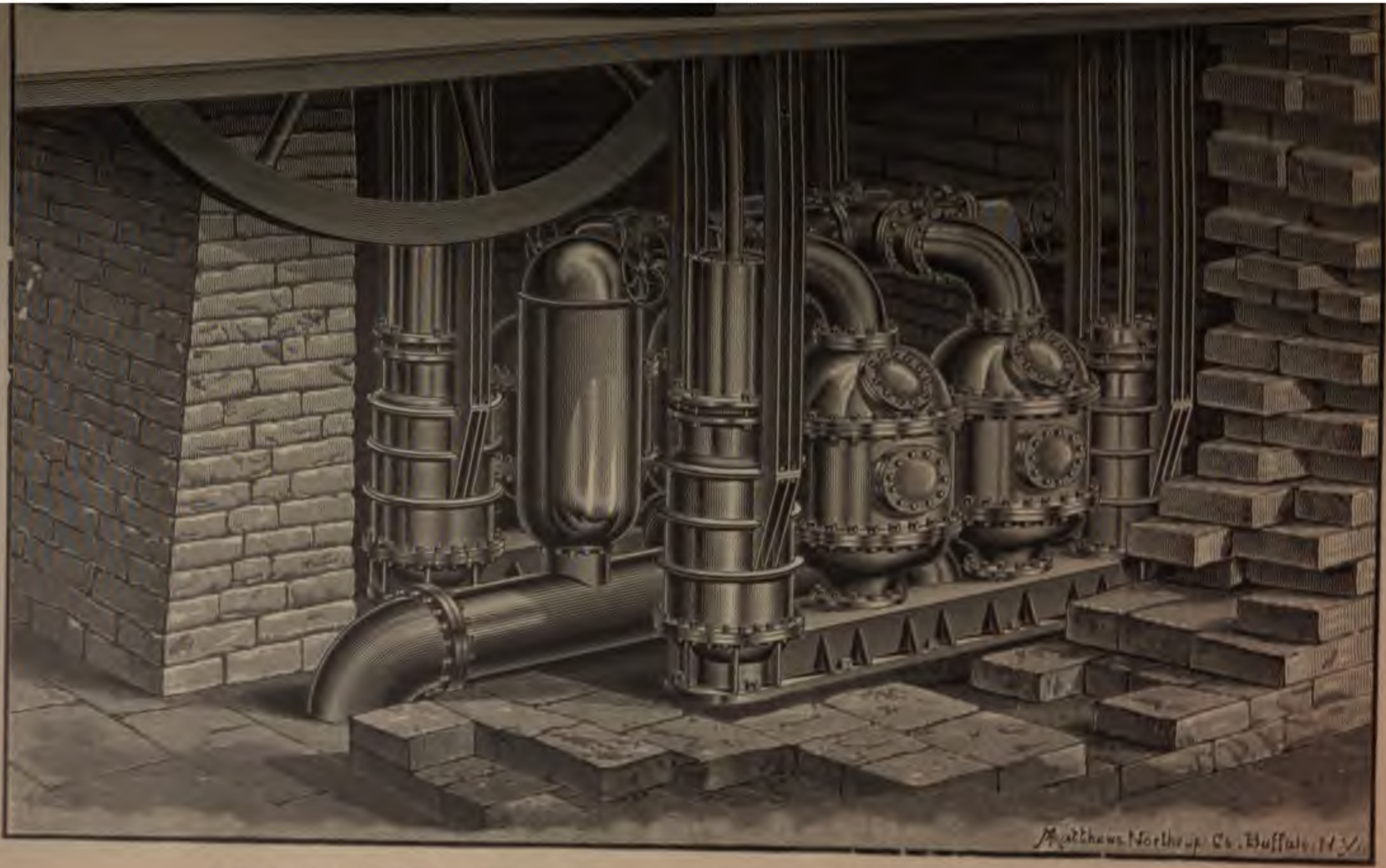

Matthews-Northrup Co., Buffalo, N.Y.

GASKILL BEAM PUMPING ENGINE, AT NASHVILLE, TENN.

PUBLIC LIBRARY

ASTOR, LENOX
TILDEN FOUNDATIONS

NASHVILLE, TENN.

FEBRUARY, 1890.

REPORT OF GEORGE REYER,

SUPERINTENDENT OF NASHVILLE WATER WORKS.

NASHVILLE, TENN., March 18, 1890.

To the Honorable Board of Public Works and Affairs:

GENTLEMEN—The following report on the duty and capacity test of engine erected at the new pumping station by the Holly Manufacturing Company is respectfully submitted.

OBJECT OF THE TEST.

The object of test was to determine whether or not the engine, erected at the new pumping station by the Holly Manufacturing Company, was capable of pumping ten million gallons of water from the Cumberland River, the source of supply, into the reservoir on Kirkpatrick's Hill, each twenty-four hours, and performing a duty of eighty-two million foot-pounds for each one hundred pounds of coal consumed on a basis of evaporation in the boilers of eight pounds of water for each pound of coal consumed, variable either way, proportionate to the variation of actual evaporation from above rate.

The basis of above-named rate of evaporation to be the raising of temperature of feed water, aided by the condenser and other appliances common to the regular working from sixty degrees F. to the requisite heat and pressure of eighty pounds, making allowance and corrections corresponding to the actual temperature of water in the river, at the time of making test.

DESCRIPTION OF MACHINERY.

The engine is a vertical compound condensing engine, having two high-pressure cylinders thirty-three inches in diameter and two low-pressure cylinders sixty-six inches in diameter, with overhead beams.

The ends of the beams are extended beyond the high-pressure cylinder to allow the insertion of a connecting rod pin.

The lower ends of the two connecting rods are attached to the cranks, which are placed at an angle of ninety degrees to each other. A fly-wheel, twenty-four feet in diameter, is placed on the main shaft, and is common to both engines.

The pumps are single-acting plunger pumps, twenty-seven inches in diameter. These pumps are placed directly under the steam cylinders, in a dry well, the distance from the base of pumps to the bed plate of engine being fifty-eight and one-half feet.

The pump rods being an extension of the piston rods, the plungers are driven directly by the pistons, and the stroke of both piston and plunger is five feet.

The condenser is a surface condenser, part of the water from the force main passing through the tubes of the condenser, and then back into the main, so that the water used for condensing is not wasted.

There are four air pumps and four feed pumps connected to and driven by the main engine.

METHOD OF TEST.

The water pumped was measured in the reservoir, allowance being made for leakage and evaporation by actual trial after the test had been completed.

Measurements were made by the City Engineer showing the capacity of the reservoir in United States gallons for each six inches of depth from 0′ to 31′.

The arrangement used for measuring the height of water in the reservoir consisted of an electric battery and bell, a light, wooden float, having a copper plate on its upper surface, a steel tape and a return wire.

This was so arranged that when the end of tape touched the copper plate, the observer was notified by the bell, and reading could be taken to within one-sixteenth of an inch.

The water furnished to the boilers was run into a wooden tank where it was carefully weighed, thence into another tank, from which it was taken by the feed pumps connected to the main engine, and forced into the boilers.

All other water pipes connecting with the boilers were disconnected, so that no water could enter the boilers except that which passed through the weighing tank.

Readings were taken every half hour of the steam pressure, water pressure and vacuum gauges, revolution counter, temperature of water in overflow from air pumps, level of water in the wet well, and temperature of the water in the tank from which it was pumped to the boilers.

An account was also kept of the coal consumed, and the ashes taken from the furnaces.

The water taken from the steam trap on the main steam pipe was run into a barrel and weighed.

A barrel calorimeter was placed on the main steam pipe between the steam trap and the engines, a test for dryness of steam made every few hours.

Indicator cards were taken daily, a full set of which I submit for inspection.

Rainfall and barometer readings for the period of test were furnished by the observer of the U. S. Signal Service.

At the beginning of the test the height of water in the boilers and the surface of water in the tank from which the water was taken by the pumps was noted, and at the end of the test the water was brought to the same level in the tank, but was found to be some higher in the boilers. The excess of water in the boilers was determined by actually weighing the water necessary to make the difference in the height of water noted, and was found to be 1,253 pounds.

It was not convenient during the test to use water from the air pump overflow, and water was taken directly from the main to fill the tank; however, the water from condenser and jackets can be used ordinarily, and in this manner the water returned to the boilers at a temperature of 111.9° F. while the water in the tank was 50.9° F.

In order to find the economy of engine, it should be credited with the quantity of steam from 50.9° F. to 80 pounds pressure, equivalent to the heat returned to boilers by feed water.

As the feed water was pumped by pumps driven by the main engine, this work should also be accredited to engine.

The heat imparted to the entire quantity of feed water (1,014,278.5 lbs.) in raising its temperature from 50.9° F. to 111.9° F. would be equal to 53,245 pounds of steam raised from 50.9° F. to eighty pounds pressure.

The feed water (1,014,278.5 lbs.) was forced into the boilers against eighty-one pounds pressure per square inch, or an equivalent head of 187 feet. The work thus done amounts to 189,670,079.5 foot-pounds.

LEAKAGE OF RESERVOIR AND PIPE LINE.

In order to ascertain the leakage of pipe line and reservoir, just after completing the test, the water was cut off above the pumps so that there could be no leaks through the pump valves, and the full pressure due to the elevation of the reservoir was maintained on the pipe line.

It was found by a measurement taken on the following day that the surface of water in reservoir had fallen at the rate of 3.63 inches in twenty-four hours, equal to a leakage of 507,401 gallons.

The water was then drawn off until only seventeen feet remained in the reservoir, when the leakage was found to be at the rate of 1.44 inches, or 206,296 gallons in twenty-four hours. The water was again drawn off, leaving only thirteen inches of water in the reservoir, when the leakage was found to be 1.52 inches, or 197,593 gallons in twenty-four hours.

From the above data, the mean leakage of reservoir and pipe line was found to be 279,896 gallons per twenty-four hours.

QUALITY OF STEAM.

The mean of twenty-one tests for dryness of steam, made by calorimeter, show the steam to be 100 per cent dry, or just at the point of saturation.

These tests were made with a barrel calorimeter, and great care was taken in reading the weights and temperatures.

This work is always liable to error by the least variation or imperfect working of the scales and thermometer used.

The water condensed in steam pipe was trapped at a point before it reached the calorimeter, and run into a barrel, where it was weighed.

The water thus collected amounted to 11,094 pounds, or 1.09 per cent. of the entire water furnished to the boilers.

CALCULATED DATA.

A—Pounds of water weighed to boilers	1,014,278.5
B—Excess of water in boilers at end of test (in pounds)	1,253.
C—Condensed water from steam pipe (in pounds)	11,094.
D—Steam used in calorimeter (in pounds)	355.5
E—Total steam used by engine A—(B+C+D)=	1,001,576.
F—Steam pressure, in pounds (gauge)	79.99
G—Vacuum in inches	25.86
H—Water pressure, in pounds (gauge)	102.78
I—Equivalent head, in feet	237.21
J—Center of gauge to surface of water in wet well, in feet	31.5
K—Total head, in feet I+J=	268.71
L—Temperature of air pump overflow	111.9° F.
M—Temperature of feed	50.9° F.
N—U. S. gallons of water used in boilers	121,616.
O—Foot-pounds of work done by feed pumps	189,670,079.5
P—Steam in pounds, raised from 50.9° F. to 80 pounds, equal to heat imparted to feed water by condenser, etc.	53,245.
Q—Barometer	30.03

RESERVOIR DATA.

R—Depth of water in reservoir at beginning of test	0 feet 17½ in.
S—Depth of water in reservoir at end of test	31 feet ⅝ in.
T—U. S. gallons of water in reservoir at beginning of test	2,274,160.15
U—U. S. gallons of water in reservoir at end of test	50,853,730.65
V—U. S. gallons of water delivered during test U—T=	48,579,570.50
W—Leakage of reservoir and pipe line during test (gallons)	1,222,970.
X—Rainfall during the test (2.11)	147,467.
Z—Total U. S. gallons pumped (V+W+N)—X=	49,776,689.5
a—Weight of one U. S. gallon, in pounds	8.34

CAPACITY.

Test started, February 27th, 2.30 P. M.

Test ended, March 3d, 11.22 P. M.

Duration of test, 4 days, 8 hours, 52 minutes.

Total gallons pumped during test	49,776,689.
Gallons pumped each 24 hours	11,365,628.

The engine was stopped from 6 A. M. until 8 A. M. on March 3d, for want of coal. No allowance was made for this either in time or for the jacket water used while engine was stopped.

LOSS BY SLIP.

Engine counter at end of test	1,001,196.
Engine counter at beginning of test	915,947.
Revolutions during test	85,249.
Revolutions per minute	13.54+
U. S. gallons per revolution	594.86
Gallons pumped by displacement	50,711,220.14
Total gallons pumped	49,776,689.5
Loss or slip	934,530.54
Per cent. of slip	1.84

DUTY.

With an evaporation of 800 pounds of water for each 100 pounds of coal consumed, considering only the water actually delivered in reservoir, with no allowance for leakage or water used in boilers, and assuming that the entire quantity of water evaporation was furnished as steam to the engine, the duty would be

$$\frac{V \times a \times K}{A - B} \times 800 = 85{,}975{,}120 \text{ foot-pounds.}$$

In order to find the duty to which the engine is truly entitled, it should be credited with all the water pumped (Z), also with the work done by feed pumps (O) and the heat imparted to the feed water (P). The duty would then be

$$\frac{(Z \times a \times K) + O}{E - P} \times 800 = 94{,}263{,}536 \text{ foot-pounds.}$$

It is now customary to represent the duty of pumping engines in foot-pounds for each 100 pounds of coal consumed, based on an evaporation of ten pounds of water for each pound of coal consumed.

The duty of this engine is therefore given based on an evaporation of ten to one, so that it can be compared with other engines based on a like evaporation,

$$\frac{(Z \times a \times K) + O}{E - P} \times 1{,}000 = \textbf{117,829,420 foot-pounds duty,}$$

based on an evaporation of 10 to 1.

BOILERS.

A battery of three boilers of the following description was used during the tests .

Length of boilers	18 feet.
Diameter of boilers	66 inches.
Number of tubes in each	59
Diameter of tubes	4 inches.
Heating surface, square feet, each boiler	1,242.
Diameter of steam pipe	10 inches.
Length of steam pipe	100 feet.
Height of stack	125 feet.
Diameter of stack (round)	78 inches.
Grate surface, each boiler, square feet	24½.

The setting is plain; the steam pipes and domes were well covered; the tops of boilers, however, have not yet been covered.

DUTY OF ENGINE AND BOILERS.

Pounds of coal used	154,438.
Temperature of feed water	50.9° F.
Steam pressure, by gauge	82.
Pounds of water evaporated	1,013,025.5
Pounds of water evaporated for each pound of coal (actual)	6.55
Equivalent evaporation from and at 212° F.	7.88
Pounds of ashes taken from furnace	23,685.
Per cent. of ashes in coal	15.2
Pounds of water evaporated for each pound of combustible	7.74
Equivalent evaporation from each pound of combustible from and at 212° F.	9.31

Duty in foot-pounds for each 100 lbs. consumed

$$\frac{(Z \times a \times K) + O}{154{,}438} \times 100 = 72{,}223{,}992.$$

During the test the water in the wet well was 46 feet above low water, but as the engine was only accredited with a head measured from the surface of water in wet well to the surface of water in reservoir, this did not affect the determination of the duty. The capacity depended entirely on the number of revolutions the engine made, whether the river is high or low.

Very respectfully,

GEORGE REYER,
Superintendent of Water Works.

NORFOLK, VA.

August, 1890.

REPORT OF W. T. BROOKE, City Engineer.

Office of W. T. Brooke, City Engineer,
Bank Street and City Hall Avenue,
Norfolk, Va., August 6, 1890.

To the Board of Street, Sewer and Drain Commissioners, Norfolk, Va.:

Gentlemen—The Holly Manufacturing Company, who have a contract with the city for erecting a sewer pump here, having completed the same, are desirous of turning it over to the city. Owing to the fact that the quantity of sewage to be pumped is a varying one and that the amount necessary to make a duty trial of the engine cannot be furnished to the pumps and consequently the requisite head to pump against cannot be obtained, we are unable to give the engine a thorough duty trial at present, and I have been compelled to simply make a running trial of the engine for capacity and general working. To do this even the sewage has had to be returned to the well and re-pumped, and as such a condition results in a continued churning of the more solid deposits in the bottom of the well, we necessarily have hesitated running the engine under such conditions any longer than absolutely necessary.

Under the above circumstances we started the inspection at 10.26 A. M. this forenoon, and completed it at 1 P. M., running the engine for two hours and thirty-four minutes. The number of revolutions made in that time were 3,750; the capacity (gross) of the pumps per revolution is 208.894 gallons, or deducting 3 per cent. in accordance with the specifications (net) 202.628 gallons, which, multiplied by the number of revolutions, gives, as having been pumped in the time specified above, 759,855 gallons, or the rate of 4934.12337 gallons per minute.

At the above rate the pumps would deliver 7,105,137.6 gallons per diem, but as the pistons were making more travel than called for by the specifications this amount has been reduced to the proper piston speed, viz., 120 feet per minute, or 24 revolutions of the crank, which is the same thing. Really the revolutions during the period of trial show an average of 24.35 per minute, and the reduction of the delivery to suit what the specifications call for, shows a capacity of 7,003,010 gallons per diem, so that it is clear that the capacity of the pump is in accordance with the requirements.

During the working the engine moved smoothly and every part seemed to perform its proper functions satisfactorily. The steam pressure shown by the gauge on the engine averaged 64 pounds, and the vacuum gauge 23½ inches. This latter is not as good a vacuum as I would like for the condenser to show, but I think it is partly caused by the high temperature of the injection water, which will decrease later on. This can only be determined certainly when the temperature of the injection is reduced to 60° or less. It may be said, however, in this connection, that all gauges are more or less uncertain, and this is rather indicated in this case from the fact that an indicator card taken from the low pressure cylinder shows an effective vacuum of 12 pounds.

This would tend to show that the vacuum gauge was wrong and that it should have read about 24½ inches, which is as good probably as could be expected under the circumstances.

The indicator diagrams show for themselves. They are almost theoretically perfect and scarcely to be improved on. They are on file at this office and can be inspected at any time.

Altogether I think the engine is ready for acceptance and recommend that it be received by your Board and the final settlement made with the contractors.

Very respectfully,
W. T. BROOKE,
City Engineer.

BUFFALO, N. Y.

October, 1890.

REPORT OF

LOUIS H. KNAPP, Engineer and Superintendent Buffalo City Water Works, and FRANK W. HOLLY, Engineer and Superintendent Holly Manufacturing Company.

DUTY AND CAPACITY TRIALS GASKILL COMPOUND PUMPING ENGINE.

Buffalo, N. Y., Water Works, October 29, 1890.

To the Honorable Board of Water Commissioners of the City of Buffalo—Louis P. Reichert, James Ryan *and* S. K. Worthington:

Gentlemen—We have the honor to report to you upon the performance of the Gaskill Compound Pumping Engine recently erected in the new addition to your lower pump house by the Holly Manufacturing Company of Lockport, N. Y., as follows:

The engine is the horizontal beam rotative type, compound condensing, and is practically two independent and complete engines coupled with cranks at quarters upon the shaft, with one fly-wheel common to both.

The high-pressure steam cylinders are placed above and parallel with the low-pressure steam cylinders, with their pistons moving in opposite directions.

Each half of the engine contains one high-pressure steam cylinder, one low-pressure steam cylinder and one double-acting plunger pump, with central packed gland.

The steam cylinders are jacketed sides and heads.

The pumps are in line with the low-pressure cylinders, with stiff connections between the low-pressure piston and the plunger. The housings which support the engine beams are placed between the pump and steam cylinders, with the bearings for beam centers midway between the center line of high and low-pressure steam cylinders.

The upper pins of the engine beams are connected by means of short links to the pins of the cross-heads of the high-pressure pistons, and the lower pins of the beam are similarly connected to the pins of the cross-heads of the low-pressure pistons.

The power developed in the high-pressure cylinders is transmitted through the engine beams to the pump plungers, and the power developed in the low-pressure cylinders is transmitted to the pump plungers direct through stiff connections.

The upper end of the beams is also connected with the fly-wheel shaft through suitable connecting rods, the pedestal of which rests on top of the pump delivery chamber.

The high-pressure inlet valves are double-beat poppets, operated by a "drop" cut-off adjustable by hand. The low-pressure steam inlet valves which are also the high-pressure exhaust valves, and the low-pressure exhaust valves are multi-ported or "gridiron" slides, worked by eccentrics from a lay-shaft lying alongside the high-pressure cylinders and driven by means of gearing attached to the fly-wheel hub.

PRINCIPAL DIMENSIONS OF ENGINE.

Diameter of H. P. cylinders	36 inches.
Diameter of L. P. cylinders	72 inches.
Diameter of H. P. piston rods	5 inches.
Diameter of L. P. piston rods	5½ inches.
Stroke of each piston	48 inches.
Clearance of H. P. cylinder	3 per cent.
Clearance of L. P. cylinder	2.5 per cent.
Diameter of pump plungers	33 inches.
Stroke of pump plungers	48 inches.
Diameter of plunger rods	6 inches.

BEAMS, CRANKS, SHAFTS AND FLY-WHEELS.

Center to center beam pins	6 feet 8½ inches.
Beam shaft, diameter at beam	14 inches.
Beam shaft, diameter at bearings	12 inches.
Cross-head pins, H. P. and L. P., diameter	7⅜ inches.
Cross-head pins, lower and upper, diameter	7¾ inches.
Crank pins (2)	9½×12½ inches.
Crank shaft, diameter at fly-wheel	17½ inches.
Crank shaft, diameter at bearings	16 inches.
Fly-wheel, diameter	23 feet 4 inches.
Fly-wheel, weight, tons	31

CONDENSER AND AIR PUMPS.

Condensers, style	Jet.
Condensers, number	2
Condensers, diameter, inches	54
Condensers, deep, inches	30
Air pumps, style	Single-acting.
Air pumps, number	4
Air pumps, diameter, inches	24
Air pumps, stroke, inches	25.5

FEED PUMPS.

Feed pumps, style	Single-acting.
Feed pumps, number	4
Feed pumps, diameter, inches	6
Feed pumps, stroke, inches	10.5

STEAM PIPES.

Steam pipe to engines	10 inches.
Exhaust pipe to condenser	14 inches.

PUMPS.

Main suction pipe, diameter, inches	36
Suction pipe, each pump, diameter, inches	30
Main discharge pipe, diameter, inches	36
Discharge pipe, each pump, inches	30
Pump valves, sets to each pump	4
Pump valves, number, suction set	306
Pump valves, number, discharge set	288
Pump valves, total one pump	1188
Ratio area through suction valve seats to area of plunger, per cent.	36.4

BOILERS.

The boilers are the horizontal return tubular type, erected in brick setting, the smoke being carried under the boilers through the tubes and back over the top to the smoke stack. They were furnished and set by the City of Buffalo, independent of the the pumping engine furnished by the Holly Manufacturing Company.

Dimensions of Boilers.

Number of boilers	4
Diameter of shell	66 inches.
Length of shell	16 feet.
Number of tubes	66
Diameter of tubes	4 inches.
Length of tubes	16 feet.
Grate surface	33 square feet.
Heating surface in boiler	1196 square feet.
Ratio of heating surface to grate surface	36.24
Chimney (8 × 8 feet square), cross section	64 square feet.
Chimney, height	127 feet.

DUTY TRIAL.

The contract requires that the engine shall develop a duty of 105,000,000 foot-pounds for every 100 pounds of coal consumed under the boilers, and shall be computed by the formula mentioned below.

The trial for duty commenced at 10.00 A. M., October 29th and ended at 10.00 P. M., October 29th. During this trial, at intervals of fifteen minutes, readings were taken of the revolution counter, steam, vacuum and water pressure gauges on engine, jacket pressure gauge, distance center water pressure gauge to water in pump-well, in engine room, and steam gauges and water levels on boilers were read every fifteen minutes.

The temperature of feed water, overflow from condenser, injection water, air in engine room, boiler room and atmosphere and barometer were read every thirty minutes.

Average water pressure by gauge	82.286 pounds.
Average height center of water pressure gauge above water in pump well	26.336 feet.
Average steam pressure gauge on engine	91.90 pounds.
Average steam pressure gauge on engine, corrected for water column	89.16 pounds.
Average vacuum by gauge	26.08 inches.
Average vacuum by gauge	12.803 pounds.

Average barometer	28.778 inches.
Average barometer	14.127 pounds.
Average temperature of feed water	124.32° F.
Average temperature of overflow	99.88° F.
Average increase in temperature of feed water by addition of jacket water	24.44° F.
Average temperature of injection water	55.92° F.
Average temperature of air, engine room	78.32° F.
Average temperature of air, boiler room	66.88° F.
Average temperature of atmosphere	45.00° F.
Average steam pressure, boilers, corrected	92.841 pounds.
Total revolutions	10,432
Total coal consumed	13,500 pounds.
Total ash weighed	1,095 pounds.
Percentage of non-combustible	8.11
Water pressure by gauge	82.286 pounds.
Distance center of gauge to water in well	11.408 pounds.
Frictional resistance through pumps (as per contract)	1.00 pound.
Total head pumped against	94.694 pounds.

$$\text{and } D = \frac{200 \times A \times P \times N \times S}{C} \text{ in which}$$

D = Duty in foot-pounds per 100 pounds of coal.
A = Aggregate area of pump plungers = 2239.926.
P = Pressure per square inch of pump plungers = 94.694.
N = Number of revolutions or double strokes of engine = 10,432.
S = Stroke of both pump plungers = 4 feet.
C = Pounds of coal consumed under boilers = 13,500,
then,

$$D = \frac{200 \times 2239.926 \times 94.694 \times 10432 \times 4}{13500} = 131,120,226.9$$

foot-pounds per 100 pounds of coal.

And duty per 100 pounds of combustible	**142,692,594.8 foot-pounds.**
Duty guaranteed on coal	**105,000,000.0 foot-pounds.**
Excess, actual duty over guarantee, or 24.87 per cent.	**26,120,226.9 foot-pounds.**
Duty of steam end per 100 pounds of coal	140,326,009

DATA FROM DIAGRAMS.

Mean effective pressure, left H. P. cylinder, front	61.50 pounds.
Mean effective pressure, left H. P. cylinder, back	60.40 pounds.
Mean effective pressure, right H. P. cylinder, front	58.925 pounds.
Mean effective pressure, right H. P. cylinder, back	59.625 pounds.
Mean effective pressure, left L. P. cylinder, front	13.853 pounds.
Mean effective pressure, left L. P. cylinder, back	13.260 pounds.
Mean effective pressure, right L. P. cylinder, front	12.486 pounds.
Mean effective pressure, right L. P. cylinder, back	12.593 pounds.
Average initial pressure, left H. P. at atmosphere	90.83 pounds.
Average initial pressure, right H. P. at atmosphere	88.92 pounds.
Average terminal pressure, left H. P. at atmosphere	26.75 pounds.
Average terminal pressure, right H. P. at atmosphere	23.92 pounds.

Average initial pressure, left L. P. at atmosphere	15.83	pounds.
Average initial pressure, right L. P. at atmosphere	15.06	pounds.
Average terminal pressure, left L. P. absolute	9.09	pounds.
Average terminal pressure, right L. P. absolute	8.90	pounds.
Effective vacuum	12.666	pounds.
Horse-power developed	797.	
Coal per I. H. P. per hour	1.411	
Duty per 100 pounds of coal, steam end	140,326,009	foot-pounds.
Total moment, steam end	226,814	pounds.
Total moment, water end	212,107.5	pounds.
Percentage of useful effect realized in work of pump	93.51	
Loss between steam and water end	6.49	per cent.
Expansion by volumes	11.172	
Expansion by pressures	11.562	
Decimal of cut-off, H. P. cylinders	0.3545	

The diameter of plungers was measured and was found to be 38 inches; the length of stroke was determined by measurement to be 48 inches. Diameter of pump plunger rod is 6 inches.

The coal burned was an excellent quality anthracite, and was taken directly from the coal bunkers of the boiler-house without being hand-picked or otherwise assorted, and was the same as used in ordinary service at the water works.

The coal was accurately weighed, in uniform charges of 500 pounds, to the boilers.

One hour before the beginning of test the fires were cleaned, and one hour before the end of test the fires were again cleaned, and were in as good condition at end as at beginning of test.

CAPACITY TEST.

Shortly after the termination of the duty trial the test for capacity was begun.

The contract formula for capacity is as follows:

$Q = 145.122 \times A \times N \times S.$
A = Sum of effective area of both plungers.
N = Mean number of revolutions per minute.
S = Stroke of pumps in feet.

In the foregoing formula three per cent. is allowed for slip or the possible imperfect filling of the pumps.

Readings were taken every fifteen minutes and the following averages were obtained:

Head pumped against, calculated by contract requirements	95.37	pounds.
Revolutions per minute	16.25	

$$Q = 145.122 \times 2239.926 \times 16.25 \times 4 = 21{,}129{,}065.16.$$

If figured from plunger displacement the capacity would be 21,782,541.4.

Capacity required by contract, 20,000,000 gallons per 24 hours.

Excess, on test, over contract requirements, 1,782,541.4, or about 5.6 per cent.

At the end of the test all bearings were found to be perfectly cool, the operation satisfactory, and the contract seems to have been more than fulfilled.

Respectfully submitted,

LOUIS H. KNAPP,
Superintendent and Engineer Buffalo Water Works.
FRANK W. HOLLY,
Engineer and Superintendent Holly Manufacturing Co.

SCHENECTADY, N. Y.

OCTOBER, 1890.

REPORT OF RICHARD M. GIFFORD, M. E.

SCHENECTADY, N. Y., October 25, 1890.

HON. E. M. JENKINS, *President Board of Water Commissioners :*

SIR—In compliance with your request, I have made a duty trial of the No. 5 Gaskill Pumping Engine, furnished by the Holly Manufacturing Company of Lockport, N. Y., and now running at the pumping station in this city. I have the honor to submit herewith my report of the same, trusting it will meet with your approval. I congratulate you upon the possession of a pumping engine which has developed such a high degree of economy, and one which its service and trial have fully demonstrated to be unequaled in operation, workmanship and finish.

Frequent examinations were made of the machinery of the engine during the trial, and at the close none of the bearings were heated; and with the exception of a slight pound in the pumps, due to some imperfect materials in the valves—and which there was no opportunity to remedy previous to the trial—the running of the engine was eminently satisfactory.

I do not hesitate to recommend the acceptance of the engine on the part of your Board, as it has more than fulfilled all the terms of the contract.

THE TEST.

I believe the trial to have been conducted in accordance with the terms of the contract, an extract from which, relating to the trial, I give as follows :

"It is agreed that within one month after the completion of the engine ready for service, duty and capacity trials of such engine shall be made, by or under the direction of the engineer, or other expert designated by the Water Commissioners, and at the expense of the party of the first part; the said trials to be made and the duty and capacity of the engine to be computed, as specified in the annexed general specification, or in this contract. During the test the contract quantity of water and the piston speed, as indicated, shall not be unnecessarily exceeded, and the water pressure will be 85 pounds per square inch, as indicated by a gauge attached to the main in the engine room, with an allowance of one pound for friction in the pumps.

"The duty will be calculated from the plunger displacement, the number of strokes and the pressure given by the gauges, with the suction lift added. A deduction of 2½ per cent. will be made for slip. In this test good, merchantable, anthracite coal will be used, to be furnished by the Water Commissioners, but may be selected by the contractor, but not hand-picked. In making this test, the fires shall, at the beginning of the test, be in their normal condition with one furnace ready for firing, the other furnace having been previously fired, in their proper order and at proper intervals of from 20 to 30 minutes respectively. Reasonably uniform intervals shall be observed in the firing throughout the test; and at its conclusion the fires shall be in the same condition as to thickness and degree of exhaustion as at the beginning."

The contract further provides that "the engine shall deliver **4,000,000** gallons of water in 24 hours, with a pressure in engine room of 85 pounds per square inch, at a piston speed not to exceed 125 feet per minute, and to show a duty of **100,000,000 foot-pounds** for each 100 pounds of coal consumed."

The trial commenced at 11 A. M., October 7th, and terminated at 11 A. M., October 8th, an uninterrupted period of 24 hours. The running of the engine was under the supervision of Mr. Charles Anderson, the representative of the Holly Manufacturing Co. The required capacity was obtained by opening a 4 inch blow-off and further regulated by the use of the hydrant opposite the pumping station.

The coal used was fair anthracite coal, furnished by the D. L. & W. R. R. Co., for general use. Stated observations were made every half hour. In the engine room the steam pressure, water pressure, vacuum, revolutions, steam jacket pressures, temperature of the overflow, and of the engine room, were noted. In the boiler room were observed the steam pressure, height of water in the boilers, reading of the meter record, temperature of the feed water, and of the water in the discharge main, and temperature of the room. Observations were made of the temperature of the external air and the height of the barometer.

The fires were brought to a normal condition at the commencement of the trial and closely maintained at that point throughout, being left in that condition at the close. The boilers used for the trial were Nos. 3 and 4, each having been in service several years. The boilers were fired alternately, the amount of coal at each firing averaging about 150 pounds. The coal was accurately weighed and brought to the boiler room in loads of 300 pounds. As soon as each full load was charged to the boilers, the amount of each was entered in the log. The firing was done at reasonably uniform periods, except at the middle of the trial, when a larger amount was charged, owing to the fires being thoroughly cleaned at that time.

The furnaces are provided with the Rogers Rocking Grate, and were lightly shaken at three periods, aside from the general cleaning mentioned above. At the commencement the ash pit doors were opened and remained so throughout the trial. The remarkably uniform steam pressure is chiefly due to the Bowker & Tripp Draft Regulator attached to the smoke flue. This was weighted to maintain the given pressure, and the maximum variation during the trial was but one pound.

After the conclusion of the test, experiments were made to ascertain the correction for reading the feed-water meter. Allowance was made for the water column on the steam gauges at the boilers, and all gauges used were tested and proper corrections were made accordingly. A subsequent test was made to determine the temperature and the amount of water discharged from the steam jackets. The jacket water is run directly to one of the four feed pumps which are operated by the machinery of the engine, and discharged into the feed pipe; the remainder of the feed water is taken from the overflow from the condensers and pumped through the meter, and into the boiler without the use of a heater. An account of all feed water wasted was kept and proper deductions made. The ash pits were thoroughly cleaned previous to the commencement, and again at the close of the trial, giving an accurate amount of the noncombustible portion of the fuel. Numerous indicator cards were taken during the trial, and they show an almost perfect distribution of steam, which is one of the many commendable features in the design of the engine.

According to the terms of the contract an allowance of 2½ per cent. for slip was stipulated. To this Mr. Anderson objected, and I think with just cause, as the plungers and rings are in excellent order, and this amount seems to be in excess of the average results obtained for this size of engine. It was finally agreed upon to allow 2 per cent. in the calculations for capacity and duty.

The general observations were made by members of the scientific class in Union College, under the direction of Prof. Brown, to whom also I am indebted for his interest manifested, and material assistance rendered. The boilers were fired by the regular firemen employed at the pumping station, to whose careful firing is largely due the success of the test. Acknowledgment is due to Chief Engineer Holly and his assistants for their careful management during the test; and to all who in the least contributed towards making this trial one of notable mention.

Appended hereto are the log of the trial, a brief description of the engine and boilers, and tables of the general data and results, which I have prepared in as concise a manner as practicable.

Desiring to thank you for your attentions during the trial, I am

Yours most respectfully,

R. M. GIFFORD.

GENERAL DESCRIPTION OF ENGINE AND BOILERS.

Diameter of high-pressure cylinders	18 inches.
Diameter of low-pressure cylinders	36 inches.
Diameter of pump plungers	17 inches.
Stroke of both steam pistons and pump plungers	32 inches.
Diameter of H. P. piston rods, steel	2⅝ inches.
Diameter of L. P. piston rods, steel	3⅛ inches.
Diameter of pump-plunger rods	3⅝ inches.
Diameter of crank shaft	9 inches.
Crank pins, steel, diameter × length	4⅝ inches × 5¾ inches.
Diameter of fly-wheel	12 feet 4 inches.
Cross-heads of cast iron with steel pins	
Number of air pumps (single acting)	4
Pumps fitted with "Troy" Valves	
Steam cylinders jacketed throughout, fed with live steam from the boilers	
Number of boilers (plain tubular)	2
Diameter of shell	60 inches.
Length of shell	16 feet.
Thickness of plates in shell (iron)	⅜ inch.
Thickness of heads	½ inch.
One dome 2′ 9″ diameter 3′ 7″ high, plate 5-16″ thick	
Number of tubes	54
Diameter of tubes	3¾ inches.
Fire grate, length × width	5 feet × 5 feet 3 inches.
Area of grate surface	26.25 square feet.
Total heating surface (about)	1060 square feet.
Ratio of heating surface to grate surface	40.38 to 1

Boilers are set to carry the products of combustion back under the boilers from the furnace; thence forward through the flues and back over the top of the boilers to the smoke pipe.

The engines are designed and built for 5,000,000 gallons capacity, excepting the plungers, which are 17″ diameter, reducing the capacity to 4,000,000 gallons, with the same pressures and piston speed as for 5,000,000 gallons. As the demand for an increased supply arises, the larger, or 19″ plungers may be placed in position with but little trouble. The same duty is guaranteed for the engine with the larger plungers as at present.

AVERAGES AND TOTALS OF DATA FOR TRIAL.

Duration of trial, hours	24.
Number of observations	49.
Number of revolutions in 24 hours	37,334.
Number revolutions per minute	25.
Piston speed in feet per minute	138.1
Average steam pressure at engine, pounds	84.5
Average steam pressure at boilers, pounds	84.6
Average vacuum, inches	28.21
Average temperature of injection, Fahr	58.1
Average temperature of overflow, Fahr	100.
Average temperature of feed water, Fahr	120.6
Average pressure of steam in steam jackets, pounds	84.5
Average pressure on discharge main in engine room per gauge, pounds	85.04
Average pressure on discharge main in engine room per gauge, in feet	196.27
Distance from center of gauge to river level, in feet	23.50
Allowance for friction in pumps, feet	2.30
Total head against pumps, feet	222.07
Discharge of pumps per revolution, gallons (calculated)	122.913
Percentage of slip in pumps (assumed)	2.
Net discharge of pumps per revolution, gallons (calculated)	120.45
Average temperature of water discharged, Fahr	58.1
Weight per gallon of water discharged at 58.1 Fahr	8.335
Delivery as per contract in 24 hours, gallons	4,000,000.
Calculated net delivery in 24 hours, gallons	4,497,057.
Excess over guarantee, per cent	12.4
Average temperature of steam jacket water discharged, Fahr	.212
Amount of steam jacket water discharged, pounds	3,456.
Total amount of water pumped through meter, cubic feet	1,088.
Less amount of water wasted, pounds	360.
Weight of one cubic foot of water at 125° (at meter), pounds	61.61
Amount of feed water to boilers, pounds	66,671.6
Total pounds of coal consumed	6,756.
Total pounds of ashes and clinkers	346.
Total pounds of combustible	6,410.
Percentage ashes and clinkers in coal	5.1
Pounds of water evaporated per pound of coal	9.8
Pounds of water evaporated per pound of combustible	10.4
Pounds of water evaporated per pound of coal from and at 212°	10.7
Average height of water above center of boilers, inches	17.625
Area of grate service in 2 boilers, square feet	52.5
Average amount of coal per hour, pounds	281.5
Average amount of coal per hour per square foot of grate area	5.3
Indicated horse power H. P. cylinders	101.29
Indicated horse power L. P. cylinders	84.06
Total indicated horse power of steam cylinders	185.35

Indicated horse power of pumps	175.45
Average amount of coal burned per H. P. per hour, pounds	1.52
Average amount of water evaporated per H. P. per hour, pounds	14.9
Duty calculated from the steam cylinder cards for 100 pounds coal	130,416,844.
Duty calculated from the steam cylinder cards for 100 lbs. combustible.	137,419,681.

The indicated horse power of the pump indicator cards is 94.6 per cent. of the indicated horse power of the steam cylinders, leaving 5.3 per cent. to overcome the friction of engine and to operate the air and feed pumps.

Average point of cut-off in H. P. cylinder, inches	8.70
Clearance, inches	.56
Total length of stroke to point of cut-off, inches	9.26
Total volume, period of admission, cubic feet	203,642.4
Equivalent weight of steam, pounds	46,837.
Weight of feed water, pounds	63,215.

During the period of admission, then, there were 16,377 pounds of water, or 25.9 per cent. of the total steam and water which entered the cylinder, leaving 74.1 per cent. of the steam accounted for at the point of cut-off.

CALCULATIONS FOR CAPACITY.

Number of revolutions in 24 hours	37,334.
Area of 17" plungers, square inches	226.981
½ area of 3⅝" rod, square inches	5.160
Mean area of plunger, square inches	221.821
Stroke of plungers, inches	32.
Displacement of pumps per revolution in cubic in., 4 × 32 × 221.821 =	28,393.088
Cubic inches in one gallon	231.

$$\text{Total gallons per revolution} = \frac{28,393.088}{231.} = 122.913.$$

Total gallons in 24 hours, 122.913 × 37,334 =	4.588,833.94
Less 2 per cent. for slip	91,776.67
Net capacity in 24 hours, gallons	4,497,057.27

Or over 12 per cent. more than guaranteed capacity.

CALCULATIONS FOR DUTY.

Total number of gallons in 24 hours (G.)	4,588,833.94
Weight of one gallon of water at 58.1° Fahr. (W.)	8.335
Total head against pumps, in feet (H.)	222.07
Total amount of coal consumed in 24 hours, pounds (C.)	6,756.
Total amount combustible in coal, pounds	6,410.

$$\text{Duty} = \frac{\overset{G}{4,588,833.94} \times \overset{W}{8.335} \times \overset{H}{222.07} \times 100}{\underset{C}{6,756}} = 125,721,106.$$

Less 2 per cent. for slip	2,514,422.
Net duty for 24 hours, foot pounds	123,206,684.
Or over 23 per cent. more than guaranteed duty of	100,000,000.
Duty calculated on 6,410 pounds combustible matter	132,507,300.
Less 2 per cent. for slip	2,650,146.
Net duty for combustible, foot-pounds	129,857,154.

Or the work of raising 129,857,154 pounds of water 1 foot high with 100 pounds of combustible consumed.

LOG OF TRIAL—ENGINES.

No.	TIME.	READING OF COUNTER.	REVOLUTION PER MINUTE.	STEAM.	VACUUM.	BAROM.	PRESSURE IN STEAM JACKETS.	TEMP. OF ENGINE ROOM.
1	11.00 A. M.	622607	....	85	28.	29.53	85	76
2	11.30	623352	24.8	85	28.	......	85	85
3	12.	624144	26.4	85	28.	......	85	87
4	12.30 P. M.	624925	26.	85	28.	......	85	89
5	1.	625711	26.2	85	28.	29.50	85	87
6	1.30	626502	26.3	85	28.	......	85	95
7	2.	627266	25.4	85	28.	......	85	88
8	2.30	628038	25.7	85	28.	......	85	89
9	3.	628838	26.6	85	28.	29.51	85	89
10	3.30	629591	25.1	85	28.	......	85	89
11	4.	630356	25.5	85	28.	......	85	90
12	4.30	631132	25.8	85	28.25	......	85	89
13	5.	631891	25.3	85	28.25	29.50	85	90
14	5.30	632644	25.1	85	28.25	......	85	86
15	6.	633386	24.7	85	28.25	......	85	88
16	6.30	634145	25.3	85	28.25	......	85	88
17	7.	634893	24.9	85	28.25	29.52	85	87
18	7.30	635630	24.5	85	28.25	......	85	87
19	8.	636387	25.2	85	28.25	......	85	87
20	8.30	637148	25.3	84	28.25	......	84	88
21	9.	637949	26.7	84	28.25	29.54	84	90
22	9.30	638742	26.4	84	28.25	......	84	89
23	10.	639532	26.3	84	28.25	......	84	89
24	10.30	640315	26.1	84	28.25	......	84	90
25	11.	641110	26.5	84	28.25	29.54	84	96
26	11.30	641889	25.9	84	28.25	......	84	90
27	12.	642675	26.2	84	28.25	......	84	90
28	12.30 A. M.	643460	26.1	84	28.25	......	84	90
29	1.	644233	25.7	84	28.25	29.40	84	90
30	1.30	645012	25.9	85	28.25	......	85	90
31	2.	645795	26.1	84	28.25	......	84	90
32	2.30	646576	26.0	85	28.25	......	85	90
33	3.	647351	25.8	85	28.37	29.37	85	90
34	3.30	648128	25.9	85	28.37	......	85	90
35	4.	648905	25.9	85	28.37	......	85	90
36	4.30	649686	26.	85	28.25	......	85	90
37	5.	650455	25.6	85	28.37	29.56	85	90
38	5.30	651256	26.7	85	28.37	......	85	90
39	6.	652038	26.1	84	28.25	......	84	90
40	6.30	652824	26.2	84	28.25	......	84	90
41	7.	653617	26.4	84	28.25	29.60	84	90
42	7.30	654406	26.3	84	28.25	......	84	90
43	8.	655172	25.5	84	28.25	......	84	90
44	8.30	655970	26.6	84	28.50	......	84	90
45	9.	656763	26.4	84	28.25	29.52	84	94
46	9.30	657565	26.7	84	28.25	......	84	93
47	10.	658341	25.8	84	28.25	......	84	92
48	10.30	659142	26.7	84	28.25	......	84	90
49	11.	659941	26.6	84	28.25	29.56	84	90
Totals		37334						
Means			25.9	84.5	28.21	29.51	84.5	89.3

LOG OF TRIAL—PUMPS.

TIME.	WATER PRESSURE.	HEAD IN FEET.	CENTER OF GAUGE ABOVE RIVER LEVEL.	TOTAL HEAD IN FEET.	TEMPERATURE OF RIVER WATER.
11.00 A. M.	85	196.18	23.50	219.68	60
11.30	85	196.18	23.50	219.68	60
12.	85	196.18	23.50	219.68	60
12.30 P. M.	85	196.18	23.50	219.68	60
1.	85	196.18	23.50	219.68	60
1.30	85	196.18	23.50	219.68	60
2.	85	196.18	23.50	219.68	60
2.30	85	196.18	23.50	219.68	60
3.	85	196.18	23.50	219.68	60
3.30	85	196.18	23.50	219.68	60
4.	85	196.18	23.50	219.68	60
4.30	85	196.18	23.50	219.68	60
5.	85	196.18	23.50	219.68	60
5.30	85	196.18	23.50	219.68	60
6.	85	196.18	23.50	219.68	60
6.30	85	196.18	23.50	219.68	59
7.	85	196.18	23.50	219.68	59
7.30	85	196.18	23.50	219.68	58
8.	85	196.18	23.50	219.68	58
8.30	85	196.18	23.50	219.68	58
9.	85	196.18	23.50	219.68	58
9.30	85	196.18	23.50	219.68	58
10.	85	196.18	23.50	219.68	58
10.30	85	196.18	23.50	219.68	58
11.	85	196.18	23.50	219.68	57
11.30	86	196.48	23.50	221.98	57
12.	86	196.48	23.50	221.98	57
12.30 A. M.	85	196.18	23.50	219.68	57
1.	85	196.18	23.50	219.68	57
1.30	85	196.18	23.50	219.68	57
2.	85	196.18	23.50	219.68	57
2.30	85	196.18	23.50	219.68	57
3.	85	196.18	23.50	219.68	57
3.30	85	196.18	23.50	219.68	57
4.	85	196.18	23.50	219.68	57
4.30	85	196.18	23.50	219.68	57
5.	85	196.18	23.50	219.68	57
5.30	85	196.18	23.50	219.68	57
6.	85	196.18	23.50	219.68	57
6.30	85	196.18	23.50	219.68	57
7.	85	196.18	23.50	219.68	57
7.30	85	196.18	23.50	219.68	56
8.	85	196.18	23.50	219.68	56
8.30	85	196.18	23.50	219.68	57
9.	85	194.18	23.50	219.68	57
9.30	85	196.18	23.50	219.68	57
10.	85	196.18	23.50	219.68	57
10.30	85	196.18	23.50	219.68	57
11.	85	196.18	23.50	219.68	57
	85.04	196.27	23.50	219.77	58.1

LOG OF TRIAL—BOILERS.

No.	Time.	Steam.	Coal Brought In.	Time Charged.	Meter Record.	Cubic Feet.	Temp. of Feed Water.	Water Level in Boilers Above Center Line.		Temp. Boiler Room.	Temperature External Air.
								No. 3.	No. 4.		
1	11.00 A. M.	85	....	....	44852	..	123	16⅜	16½	74	52
2	11.30	85	....	....	44882	30	119	16¾	16	76	52
3	12.	85	300	11.45 A. M	44907	25	119	18½	18½	84	52
4	12.30 P. M.	85	....	....	44929	22	123	18¾	18¾	96	52
5	1.	85	300	1. P. M.	44952	23	124	19	17	84	52
6	1.30	85	300	1.35	44974	22	124	19	18	88	52
7	2.	85	....	....	44996	22	124	18¾	16¾	95	52
8	2.30	85	....	....	45018	22	124	18½	17	97	52
9	3.	85	300	3.	45032	14	125	18½	16¼	99	51
10	3.30	85	....	....	45063	31	123	18½	16¾	98	50
11	4.	85	300	4.	45085	22	123	18½	16	99	50
12	4.30	85	....	....	45108	23	124	17	17½	99	50
13	5.	85	300	5.10	45129	21	123	17¼	17½	96	49½
14	5.30	85	....	....	45150	21	122	17½	16¼	92	49
15	6.	85	300	6.10	45172	22	121	17	17¼	97	49
16	6.30	85	....	....	45191	22	122	17½	17¼	98	49
17	7.	85	300	7.07	45216	22	121	16½	17½	93	48½
18	7.30	85	....	....	45237	21	122	17½	16¼	93	48½
19	8.	85	300	8 20	45259	22	120	17	17¼	93	48½
20	8.30	84	....	....	45280	21	118	16¾	16	99	49
21	9.	84	....	....	45304	24	123	16¾	17	95	49
22	9.30	84	300	9.35	45327	23	124	17¼	16½	98	48
23	10.	84	....	....	45355	28	122	17½	17¼	98	48
24	10.30	84	300	10.45	45377	22	120	17½	17¼	89	48
25	11.	84	....	....	45401	24	120	17½	17½	90	48
26	11.30	84	....	....	45425	24	120	17¾	17¾	92	48
37	12.	84	300	11.45	45449	24	120	18	17¾	94	48
28	12.30 A. M.	84	300	12.15 A. M.	45473	24	119	17¾	18	95	48
29	1.	84	300	1.15	45497	24	120	18	18	97	48
30	1.30	85	....	....	45520	23	119	18	18	98	48
31	2.	84	300	2.20	45544	24	120	18¼	18¼	98	48
32	2.30	85	....	....	45567	23	121	18	18½	97	48
33	3.	85	....	....	45589	22	119	18	18½	99	49
34	3.30	85	300	3.30	45611	22	119	18¼	18¼	84	49
35	4.	85	....	....	45634	23	118	18	18½	90	49
36	4.30	85	300	4.30	45657	23	119	18	18½	98	49
37	5.	85	....	....	45679	22	118	19	17½	98	49
38	5.30	85	300	5.20	45702	23	119	18½	18	93	49
39	6	85	...	....	45725	23	118	18	18	98	49
40	6.30	84	300	6 20	45748	23	117	18	18	96	49
41	7.	84	....	....	45770	22	118	18	18¼	98	50
42	7.30	84	300	7.20	45794	24	119	18	18	96	50
43	8.	84	....	....	45816	22	117	18	18	88	52
44	8 30	84	300	8.25	45839	23	119	18	18¼	95	53
45	9.	84	....	....	45863	24	119	18	18½	86	53
46	9.30	85	....	....	45877	14	117	17	17	88	53
47	10.	85	300	10.	45895	18	122	16½	17	96	53
48	10.30	85	....	....	45918	23	120	17	16¾	86	54
49	11.	85	156	10.45	45940	22	120	16½	16½	86	54
Totals			6756		1088						
Means		84.6				22.66	120.61	17¾	17½	93. 1	49.9

COVINGTON, KY.

NOVEMBER, 1890.

PRELIMINARY REPORT OF G. BOUSCAREN,

CONSTRUCTING ENGINEER.

CINCINNATI, December 11, 1890.

Trustees Covington Reservoir, Covington, Ky.:

GENTLEMEN—I beg to submit the following report on the duty test recently made of the Covington pumping engines, built by the Holly Manufacturing Co., under contract with your Board.

The general specifications under which proposals were received for these pumping engines stipulate that "each pumping engine, working either singly or jointly with the normal pressure of 100 lbs. of steam in the boilers, shall deliver water in the reservoir at the rate of 5,000,000 United States standard gallons in 24 hours, with a consumption of coal not exceeding 18,000 lbs. in 24 hours, when the level of water in the river is 385 feet below the center of the force pipe at the apex on the hill. The consumption shall be proportionally less as the water in the river rises above the datum referred to.

"After a period of ninety days operation the final tests for duty shall be made in the following manner:

"1st. Both pumping engines shall be worked simultaneously, in normal condition with a steam pressure of 100 lbs. to the square inch in the boilers, and a rate of delivery into the reservoir not less than 10,000,000 gallons in 24 hours for a period of not less than 48 hours.

"2d. A similar test shall be made with each engine working singly, in normal condition with 100 lbs. steam pressure in the boilers. The coal consumed shall be carefully weighed. The quantity of water shall be carefully measured over a weir at the reservoir, and an accurate record shall be kept of the elevation of water in the river during the test."

In the contract, as executed by the Holly Manufacturing Co., these conditions were slightly modified by the following clause:

"The above described machinery will be guaranteed to be capable of pumping 5,000,000 United States gallons of water per 24 hours against a static head of 378.1 feet, at a piston speed of 120 feet per minute with 90 lbs. steam, and to develop a duty equivalent to rising 2,500,000 gallons 378.1 feet high in 24 hours with a consumption of 9,000 lbs. of coal, and to raising 5,000,000 gallons 378.1 feet high in 24 hours with a consumption of 18,000 lbs. of coal, test to be conducted as per your specifications."

The changes made from the general specifications by this clause of the contract are:

1st. That a maximum static head of 378.1 feet is substituted for 385 feet, the former being the difference in elevation between extreme low water in the river and the highest point on the force pipe line at the open stand-pipe in tunnel.

2d. That a pressure of 90 lbs. of steam (inferred to apply to the working pressure at the engines, but not expressly defined) is substituted in place of 100 lbs. pressure at the boilers.

3d. That the piston speed of the engines is limited to 120 feet per minute, whereas no limit is stipulated in the general specifications. The weir used for gauging the quantities of water delivered was erected at the inlet of the north basin of the reservoir.

The tests were made in the following order :

1st. Engine No. 1, running alone, from November 25th, 12.30 P. M., to November 27th, 12.30 P. M.

2d. Engine No. 2, running alone, from November 27th, 6 P. M., to November 29th, 6 P. M.

3d. Engines Nos. 1 and 2, running together, from December 3d, noon, to December 5th, noon.

The interval of time between test No. 2 and test No. 3 was rendered unavoidable by the necessity of waiting for the necessary room in the two basins of the reservoir to receive the water pumped during the third test, and by the absence of Mr. Hermany, who was compelled to attend to urgent business in Louisville.

The water delivered during test No. 1 for the 46 hours, beginning at 1.50 P. M., November 25th, and ending at 11.50 A. M., November 27th, was carefully measured in the basin to serve as a check to the gauging at the weir. The same was done during test No. 3 for the first 24 hours run, from noon, December 3d, to noon, December 4th.

As a further check to show the change in the condition of the engines, which might have taken place during the test, short runs were made at the end of test No. 3, with each engine working alone.

For engine No. 1, from 12.15 P. M. to 1.45 P. M.

For engine No. 2, from 2.15 P. M. to 3.45 P. M.

The results of my calculations, from all the observations taken, are tabulated in the accompany exhibit.

The notes have been handed by me to Mr. Hermany, who will make his own computation and submit his report to your board next week.

The weir and basin measurements in the first test were found to agree within about one per cent., which is as close as could be expected.

In the third test the basin measurement exceeded the weir measurement by more than 3½ per cent., which can only be explained by the fact that the current in the weir trough was disturbed to such an extent by the large quantity of air carried by the water from the stand-pipe as to impair the accuracy of the gauging. The quantity of water given by the basin measurement less 0.01092 was taken as a basis for the computation of the allowable coal in that test.

My conclusion, with regard to the test, is that the engines have complied with the conditions of the contract except in the following particulars :

1st. That the pressure of steam corresponding to the specified duty is 100 lbs., instead of 90 lbs.

2d. That the piston speed necessary for the delivery of the specified quantity of water in 24 hours exceeds the specified speed by 1½ to 2½ per cent.

These discrepancies being purely of a technical nature, I recommend the acceptance of the engines from the contractors' hands.

Very respectfully,

G. BOUSCAREN,

Const. Engineer

EXHIBIT

SHOWING RESULTS OF DUTY TEST OF COVINGTON PUMPING ENGINES.

	Engine No. 1.	Engine No. 2.	Engine Nos. 1 and 2 Combined.	Engine No. 1, Short Run.	Engine No. 2, Short Run.
Water delivered (weir measurement)	10,909,269 gals.	10,937,261 gals.	21,546,323 gals.	50,245 c. f.	47,806 c. f.
Water delivered (basin measurement)	*10,570,610 gals.	Not taken.	†11,153,805 gals.	Not taken.	Not taken.
Discrepancy between weir and basin (basin in excess of weir)	For 46 hours 0.01092 per ct.	....	For 1st 24 hours 0.03507 per ct.	....	....
Number of revolutions of engine	63,795	64,299	No. 1—64,899 No. 2—65,295	2230	2106
Displacement of pumps	11,275,432 gals.	11,364,510 gals.	23,011,103 gals.	52,689 c. f.	49,759 c. f.
Slip of pumps by weir measurement	0.0325 per ct.	0.0376 per ct.	0.0637 per ct.	0.0464 per ct.	0.0493 per ct.
Slip of pumps by basin measurement	0.0217 per ct.	Not taken.	0.0308 per ct.	....	....
Average depth surface of water in well below floor engine house	49.279 feet.	54.136 feet.	60.330 feet.		
Depth of water below center of force main at stand-pipe	0.125 ft. below	0.125 ft. below	0.375 ft. above		
Total average lift	356.964 feet.	361.821 feet.	368.515 feet.		
Quantity of water assumed in computation of allowable coal	10,909,269 gals.	10,937,261 gals.	22,061,091 gals.		
Allowable coal	37,078 lbs.	37,679 lbs.	77,406 lbs.		
Coal burnt	35,041 lbs.	36,675 lbs.	76,725 lbs.		
Margin of allowable coal over coal burnt	2,037 lbs.	1,004 lbs.	681 lbs.		
Margin in per cent	0.055	0.0266	0.0088		
Actual piston speed	132.9 feet.	134 feet.	No. 1—134.9 ft. No. 2—135.6 ft.		
Piston speed on basis of 5,000,000 gallons for each engine	121.8 feet.	122.5 feet.	Average 122.9 ft.		
Excess of piston speed	0.015 per ct.	0.0204 per ct.	0.024 per ct.		

The weights of coal allowable is given by the following formula, in which

H = total average lift.

Q = quantity of water delivered in gallons.

For tests Nos. 1 and 2,

$$W = 18000 \times \frac{H \times Q}{378.1 \times 5}$$

For test No. 3,

$$W = 36000 \times \frac{H \times Q}{378.1 \times 10}$$

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* For the same 46 hours weir measurement = 10,456,400 gallons.

† For the same 24 hours weir measurement = 10,775,952 gallons.

COVINGTON, KY.

NOVEMBER, 1890.

FINAL REPORT OF CHAS. HERMANY, CONSULTING ENGINEER.

LOUISVILLE, KY., December 18, 1890.

To the Trustees of the Covington Reservoir, Covington, Ky.:

GENTLEMEN—Below please find my report upon the fuel and capacity tests made with two pairs of compound steam pumping engines, built by the Holly Manufacturing Company for the Covington, Ky., Reservoir Trustees.

ENGINE NO. 1.

1. Duration of test from 12.30 P. M., November 25th, to 12.30 P. M., November 27th, 1890—48 hours.

2. Two boilers fired during test, Nos. 3 and 4.

3. Coal burnt in 48 hours, 34,990 pounds.

4. Quality of fuel used was good Pittsburgh lump coal, forked and free from slack. The contract says: "The coal used shall be the run of the mine, ordinary quality Pittsburgh coal."

5. The average pressure of steam in boilers during test was 104.18 pounds per square inch. Pressure stipulated in contract was 90 pounds per square inch.

6. Engine No. 1 operates four single-acting plunger pumps, each 19 inches in diameter and 36 inches stroke.

7. Plunger displacement per revolution, equals

$$19^2 \times 0.7854 \times 36 \times 4 - 1.231 = 176.7456 \text{ U. S. gallons.}$$

8. Number of revolutions during test 63,795.

9. Piston speed 120 feet per minute as named in contract for 10,000,000 gallons in 48 hours.

10. Average piston speed during test was

$$\frac{63{,}795 \times 6}{48 \times 60} \times \frac{10{,}000{,}000}{11{,}030{,}201} = 120.49 \text{ feet per minute.}$$

11. Volume of water pumped during 48 hours, estimated by plunger displacement was, 63,795 × 176.7456 = 11,275,486 gallons.

12. Static head against pumps from low water in the river to water surface at apex at force main at stand-pipe, 378.1 feet as specified.

13. Average vertical distance during test from engine-room floor to water in well 49.342 feet.

14. Average height during test from engine-room floor to water surface at apex in force main at stand-pipe 307.81 feet by engineer's levels. Same by pressure gauge 314.61 feet.

15. Total static head against pumps during test 357.152 feet by engineer's level.

16. The mean head on the weir during test from 576 readings of hook-gauge, was 0.64405 feet; head due velocity of approach 0.0024 feet; and total head on weir 0.64735 feet. Length of weir 4.995 feet.

17. Volume of water delivered into reservoir in 48 hours by weir measurement (computed by Francis' well-known formula):

$$Q = 3.33 (L — 0.1 n H) H^{\frac{3}{2}}) \text{ was}$$

$$3.33 (4.995 — 0.1 \times 2 \times 0.64735) \times 0.64735^{\frac{3}{2}} \times 7.4805 \times 60 \times 60 \times 48 =$$

10,908,310 gallons.

18. Volume of water delivered into reservoir in 46 hours by basin measurement.

Water in N. basin November 25th, at 1.50 P. M., by table of reservoir capacity	12,689,340 gallons.
Water in same November 27th, at 11.50 A. M	23,259,950 gallons.
In 46 hours	10,570,610 gallons.
Equivalent to	$\frac{48}{46}$
	11,030,201 gallons

in 48 hours.

19. These three different methods of determining the volume of water delivered into the reservoir, as shown in sections 7, 11, 16, 17 and 18 of this report, give the following as delivery in 48 hours, viz.:

By pump plunger displacement	11,275,486 gallons.
By weir gauging	10,908,310 gallons.
By basin measurement	11,030,201 gallons.

The lost action (slip) of the pumps was therefore 0.03366 when estimated by weir gauging, and 0.02223 compared with basin measurement

The volume of water delivered into reservoir, as determined by basin measurement, is taken as the closest approximation to accuracy, and, therefore, Engine No. 1 is given credit for having delivered 11,030,201 gallons of water into the reservoir in 48 hours.

20. The contract stipulated that the engine shall pump 10,000,000 gallons of water in 48 hours from low water level in the river to the height of the center of the force pipe at the stand-pipe (a height of 378.1 feet) with 36,000 pounds of coal, or the same volume of water with a proportionately less weight of coal as the water in the river rises above the low water level. Which alternative means, that if the conditions named in the contract do not exist at the time of making the test, then the result of the test made under changed conditions, shall be the *mechanical equivalent* of the work stipulated in the contract.

Instead of pumping the stipulated volume, as above stated, this engine pumped 11,030,201 gallons of water in 48 hours, 357.152 feet high with 34,990 pounds of coal, which is the mechanical equivalent of the work stipulated to be performed, with 2,619 pounds coal less than the weight allowed in the contract—a reduction of $7\frac{2}{10}$ per cent. expressed as follows, viz.:

$$\frac{11{,}030{,}201 \text{ gallons} \times 357.152 \text{ feet head}}{10{,}000{,}000 \text{ gallons} \times 378.1 \text{ feet head}} \times 36{,}000 \text{ pounds coal} — 34{,}990 \text{ pounds coal} =$$

2,519 pounds coal less than the contract allowance.

ENGINE NO. 2.

1. Duration of test 6.00 P. M., November 27th, to 6.00 P. M., November 29th, 1890, 48 hours.

2. Two boilers fired during test, Nos. 3 and 4.

3. Coal burnt in 48 hours, 36,777 pounds.

4. Quality of fuel used was good Pittsburgh coal, containing a large percentage of slack. The contract says: "The coal used shall be the run of the mine, ordinary quality Pittsburgh coal."

5. The average pressure of steam in boiler during the test was 103.81 pounds per square inch. Pressure stipulated in contract is 90 pounds per square inch.

6. Engine No. 2 is a duplicate of No. 1, and operates four single-acting plunger pumps, each 19 inches in diameter and 36 inches stroke.

7. Plunger displacement per revolution, equals

$$19^2 \times .7854 \times 36 \times 4 \times \tfrac{1}{231} = 176.7456 \text{ U. S. gallons.}$$

8. Number of revolutions during test 64,299.

9. Piston speed 120 feet per minute as named in contract.

10. Average piston speed during test was

$$\frac{64{,}299 \times 6}{48 \times 60} \times \frac{10{,}000{,}000}{10{,}926{,}128} = 122.49 \text{ feet per minute.}$$

11. Volume of water pumped during 48 hours, estimated by plunger displacement, was 64,299 revolutions × 176.7456 gallons = 11,364,565 gallons.

12. Static head against pumps from low water in the river to water surface at apex in force main at stand-pipe, 378.1 feet as specified in contract.

13. Average vertical distance during test from engine-room floor to water in well 54.115 feet.

14. Average height during test from engine-room floor to water surface at apex in force main at stand-pipe 307.81 feet by engineer's levels. Same by pressure gauge 314.61 feet.

15. Total static head against pumps during test was 361.925 feet by engineer's levels.

16. The mean head on the weir during the test from hook-gauge readings taken at regular intervals of five minutes apart was 0.64607 feet; head due velocity of approach 0.0024 feet; and total head on weir 0.64847 feet. Length of weir 4.995 feet.

17. Volume of water delivered into reservoir in 48 hours by weir measurement, computed by Francis' formula, was

$$3.33\,(4.995 - 0.1 \times 2 \times 0.64847) \times 0.64847^{\frac{3}{2}} \times 7.4805 \times 60 \times 60 \times 48 =$$

10,936,128 gallons.

18. Volume of water delivered into reservoir in the test of this engine was not measured in the basin.

19. The two different methods of determining the volume of water delivered into the reservoir give the following as the delivery in 48 hours, viz.:

By pump plunger displacement	11,364,565 gallons.
By weir gauging	10,936,128 gallons.

The result of the test made with Engine No. 1 gave the volume of water delivered, by basin measurement, as 1.01117 times that determined by weir gauging. For the reason assigned in the report on the test with Engine No. 1, the weir volume of the delivery by Engine No. 2, is multiplied by 1.01117, and the engine given credit for delivering 11,058,328 gallons water into the reservoir in 48 hours.

The lost action (or slip) of the pumps was therefore 0.03917, estimated by weir gauging, and 0.02769 compared with *estimated* basin measurement.

20. The contract requires Engine No. 2 to perform the same work with the same consumption of coal as stipulated for Engine No. 1. Engine No. 2 pumped 11,058,328 gallons of water, 361.925 feet high in 48 hours, with 36,777 pounds of coal, which is the mechanical equivalent of the work stipulated to be performed with 1,330 pounds of coal less than the weight allowed in the contract—a reduction of $3\frac{61}{100}$ per cent., expressed as follows, viz.:

$$\frac{11{,}058{,}328 \text{ gallons water} \times 361.925 \text{ feet head}}{10{,}000{,}000 \text{ gallons water} \times 378.1 \text{ feet head}} \times 36{,}000 \text{ pounds coal} - 36{,}777 \text{ lbs. coal} =$$

1,330 pounds coal less than the contract allowance.

The difference in the coal consumption by the two engines is 1,189 pounds, and is evidently due to the difference in the condition of the coal.

ENGINES NOS. 1 AND 2 WORKING SIMULTANEOUSLY.

1. Duration of test from 12 noon, December 3d, to 12 noon, December 5th, 1890—48 hours.

2. Four boilers fired during test, Nos. 1, 2, 3 and 4.

3. Coal burnt in 48 hours, 76,460 pounds.

4. Quality of fuel used was good Pittsburgh coal, containing a large percentage of slack. The contract says: "The coal used shall be the run of the mine, ordinary quality Pittsburgh coal."

5. The average pressure of steam in the boilers during the test was 103.78 pounds per square inch. Pressure stipulated in contract is 90 pounds per square inch.

6. The two engines working simultaneously operate eight single-acting plunger pumps, each 19 inches in diameter and 36 inches stroke.

7. Combined plunger displacement by the two engines, per joint revolution, equals

$$19^2 \times .7854 \times 36 \times 4 \times 2 \times \tfrac{1}{231} = 353.4912 \text{ U. S. gallons.}$$

8. Number of revolutions during test for both engines was

(64,899 for No. 1 + 65,295 for No. 2) = 130,194.

9. Piston speed 120 feet per minute as named in contract for 20,000,000 gallons in 48 hours.

10. Average piston speed during test was

$$\frac{130{,}194 \times 6}{48 \times 60} \times \frac{20{,}000{,}000}{22{,}307{,}610} \times \frac{1}{2} = 121.53 \text{ feet per minute.}$$

11. Volume of water pumped during 48 hours, estimated by plunger displacement, was 65,097 × 353.4912 = 23,011,217 gallons.

12. Static head against pumps from low water in the river to water surface at apex in force main at stand-pipe is 378.1 feet as specified in contract.

13. Average vertical distance during test from engine-room floor to water in well was 60.333 feet.

14. Average height from engine-room floor to water surface at apex in force main at stand-pipe 307.81 feet by engineer's levels. Same by pressure gauge 314.61 feet.

15. Total static head against pumps during test 368.143 feet by engineer's levels.

16. The mean head on the weir during the test, from hook-gauge readings taken at regular intervals of five minutes apart, was 1.02124 feet; head due velocity of approach 0.0087 feet; and total head in weir 1.02994 feet. Length of weir 4.995 feet.

17. Volume of water delivered into reservoir in 48 hours by weir measurement, computed by Francis' formula, was

$$3.33\,(4.995 - 0.1 \times 2 \times 1.02994) \times 1.02994^{\frac{3}{2}} \times 7.4805 \times 60 \times 60 \times 48 =$$
21,546,782 gallons.

18. Volume of water delivered into reservoir in 24 hours by basin measurement:

Water in N. basin December 3d, at 11.50 A. M ....14,616,590 gallons.
Water in same at 11.50 A. M., December 4th ....25,770,395 gallons.

In 24 hours ....11,153,805 gallons.
Equivalent to........................ $\frac{48}{24}$

In 48 hours ....22,307,610 gallons.

19. The volume of water delivered into the reservoir in 48 hours by the two engines when worked simultaneously is, therefore, as follows, viz.:

By plunger displacement ....23,011,217 gallons.
By weir gauging ....21,546,782 gallons.
By basin measurement........................22,307,610 gallons.

The lost action (slip) of the pumps was therefore 0.06796 when estimated by weir gauging, and 0.03154 when compared with the basin measurement.

The basin measurement, for reasons heretofore stated, is taken as the closest approximation to the correct volume of water delivered into the reservoir, and the two engines running jointly are therefore credited with having delivered 22,307,610 gallons of water in 48 hours.

20. The contract stipulates that the two engines running jointly shall pump 20,000,000 gallons of water in 48 hours 378.1 feet high with 36,000 pounds of coal.

The two engines running jointly did pump 22,307,610 gallons of water in 48 hours 368.143 feet high with 76,460 pounds of coal, which is the mechanical equivalent of the work stipulated to be performed with 1,732 pounds of coal less than the weight allowed in the contract—a reduction of $2\frac{27}{100}$ per cent., expressed as follows, viz.:

$$\frac{22{,}307{,}610 \text{ gallons water} \times 368.143 \text{ feet head}}{20{,}000{,}000 \text{ gallons water} \times 378.1 \text{ feet head}} \times 72{,}000 \text{ pounds coal} - 76{,}460 \text{ lbs. coal} =$$
1,732 pounds less than the contract allowance.

The results from these several tests show that the pumping engines meet the contract requirements fully (exceed them in some respects), with two exceptions, viz.: steam pressure in boilers and piston speed per minute.

These exceptions are technical when considered in connection with the entire plant and of too little importance to be made a subject for discussion.

From the tests made and a careful inspection of the whole plant, I am justified in saying that it is first-class in every respect and fulfills the contract.

Very respectfully,

CHAS. HERMANY,
Civil Engineer.

TERRE HAUTE, IND.

FEBRUARY, 1891.

PRELIMINARY REPORT OF PROF. C. L. BROWN.

ROSE POLYTECHNIC INSTITUTE,
TERRE HAUTE, IND., February 24, 1891.

The Terre Haute Water Works Company:

GENTLEMEN—In response to the request of your superintendent, Mr. Williamson, I, on February 11th, made a duty test of your new No. 4 Gaskill Pumping Engine, recently erected by the Holly Manufacturing Co., of Lockport, N. Y., at your pumping station in this city. Previous to the test the engine was examined and measurements made of its various parts, to determine how closely they complied with the specifications contained in your printed contract with the manufacturers. These were all found to be substantially as specified.

The test began at 1.30 P. M., Wednesday, February 11th, and continued without break or interruption from any cause till 1.30 A. M., Thursday the 12th, making a continuous run of twelve hours.

The following is a copy of the guarantee under the conditions of which the test was made:

The above-described machinery will be guaranteed to be capable of pumping ,000,000 United States gallons of water per 24 hours against a head of 60 pounds for lomestic service, or against a head of 125 pounds for fire service, at a piston speed not o exceed 110 feet per minute pump plunger displacement.

DUTY.

The engine will be guaranteed to perform the duty of 100,000,000 foot-pounds when pumping at its rated capacity, against a head of 60 pounds based upon an evaporation by the boilers of 10 pounds of water per pound of coal, from and at the temperature of feed water with a steam pressure of   pounds per square inch at the throttle valve when running.

CAPACITY.

Calculated from plunger displacement by terms of contract:

Number of pump plungers (single acting)	4
Average diameter pump plungers, inches	22
Average length of stroke, feet	3
Average area of pump plungers, square inches	380.1336
Average displacement pump plunger, cubic inches	13684.8096
Total displacement 4 pump plungers, cubic inches	54739.2384
Number of gallons per stroke	236.9664
Piston speed, feet per minute	110
Equivalent strokes per minute	18⅓
Equivalent strokes per hour	1100
Equivalent strokes per day, 24 hours	26400
Gallons per minute	4,344.384
Gallons per hour	260,663.04
Gallons per day, 24 hours	6,255,912.96
Contract capacity	6,000,000.00
Excess above contract capacity	255,912.96
or	4.26%
During the trial of 12 hours duration the engine counter registered revolutions	14,762
Equivalent revolutions per hour	1,230.2
Revolutions per minute	20.503
Piston speed, feet	123.02

At the above speed the amount of water pumped was as follows:

Number of gallons per stroke	236.9664
Total gallons pumped, 12 hours	3,498,098
Equivalent gallons pumped, 24 hours	6,996,196

against total head of 72.76 pounds pressure.

No fire service test of the engine was considered necessary, its ability to pump at contract speed and head having been repeatedly demonstrated in actual daily service.

By the terms of the contract, the duty should be computed on the basis of "evaporation by the boilers of 10 pounds of water per pound of coal," or, in other words, that 100 pounds of coal, the usual basis for duty calculation, should be considered as evaporating 1,000 pounds of water to dry steam. The dryness of the steam was measured by Professor Thomas Gray, with a new form of superheating calorimeter, designed by himself and built for the occasion.

The steam supplied by the boilers was of good quality, containing only 1.6 per cent. of entrained water.

For the computation of duty we have the following data, from the various readings made during the test:

Number of single-acting plungers	4
Area of single plunger, square inches	380.1336
Length of stroke, feet	3
Number of strokes during test	14,762

Height of water pressure gauge above level of water in pump-well, feet	46.61
Pressure due to height at 46.61, pounds	20.00
Average pressure by gauge, pounds	52.28
Correction, pounds	.28
Total pressure pumped against, pounds	72.76
Total water weighed to boilers, pounds	46,779
Total water used in calorimeter test, pounds	900
Total entrained water 1.6 per cent., pounds	734
Net water charged engine, pounds	45,145

$$\text{Duty} = \frac{\text{Area plunger} \times 4 \times \text{stroke} \times \text{revolutions} \times \text{pressure}}{\dfrac{\text{Net water}}{1000.}}$$

$$\text{Equals } \frac{380.1336 \times 4 \times 3 \times 14.762 \times 72.76}{45.145}$$

equals 108,328,984.29 foot-pounds per 1,000 pounds steam.

Computed by the terms of contract, the actual duty exceeds the guarantee by 8,528,984.29 foot-pounds or 8.5 per cent.

During the entire test of 12 hours duration, the engine ran smoothly without perceptible heating or undue vibration with piston speed and against a head far in excess of the contract specification. The material and workmanship were apparently satisfactory, and I am of the opinion that the pumping engine has met the requirements of specification and guarantee.

Much data was collected during the tests from numerous readings and indicator cards taken at stated intervals.

These will be tabulated and various calculated results appended, which will be presented later on in a fully detailed report.

Respectfully submitted,

C. S. BROWN.

KALAMAZOO, MICH.

FEBRUARY, 1891.

PRELIMINARY REPORT OF PROF. M. E. COOLEY.

ANN ARBOR, MICHIGAN, February 28, 1891.

To the Honorable, The Common Council, Kalamazoo, Michigan — The Committee on Fire and Water:

GENTLEMEN—I have the honor to submit a brief preliminary report on the performance of the new pumping engine recently erected at your City Water Works and tested January 6, 7, 8 and 9, 1891.

This engine, known as the Gaskill Vertical Compound Condensing Engine, was built and erected by the Holly Manufacturing Company of Lockport, N. Y., and is almost an exact duplicate of the Gaskill engine erected for your city in 1885–86. A full description of the engine will appear in my full report.

In the specifications and contract between the City of Kalamazoo and the Holly Manufacturing Company, this engine is "guaranteed to be capable of lifting 3,000,000 United States gallons of water per 24 hours, pump plunger displacement, against a head of 120 feet for domestic, and against a head of 125 pounds for fire service, at a piston speed of 135 feet per minute."

Further, "the engine will be guaranteed to perform a test duty on a run of 24 hours of 95,000,000 foot-pounds per 100 pounds of best coal, when pumping at its rated capacity against a head of 120 feet, with a steam pressure of 75 pounds per square inch at the throttle when running."

It gives me much pleasure to inform you that the requirements of the contract, as above stated, have been fully complied with, and that you possess in this new engine a machine capable of performing the duties required of it easily, safely and economically. This engine works with much smoothness, even under the hardest service imposed upon it, and appears to be a thoroughly well built machine.

The City of Kalamazoo is to be congratulated on possessing a pair of engines which have so fully satisfied the requirements in all particulars ; and is entitled to feel a just pride in its City Water Works, which is perhaps one of the most unique, well arranged and complete systems in the country.

The results secured in the trial tests of January 6th, 7th, 8th and 9th, are as follows (Note—These results are subject to further correction for instrumental errors):

CAPACITY TEST.

The contract provides for no allowance for *slip*, and stipulates that the capacity shall be figured from the pump plunger displacement at a piston speed of 135 feet per minute. The result thus figured is 3,174,371.4 United States gallons. The requirements of 3,000,000 United States gallons is exceeded by 174,371.4 gallons, equal to 5.8 per cent., which is considerable in excess of any probable slip for this class of pumps. The piston speed for 25.25 hours averaged 135.05 feet per minute, and the head by gauge pumped against 124.8 feet, the steam pressure at the throttle being 75.8 pounds per square inch.

FIRE TEST.

During the fire test, which was of 20 minutes duration, a pressure of 127.14 pounds per square inch was maintained in the force main at a piston speed of 139.75 feet per minute, and with a steam pressure of 74.5 pounds per square inch.

The engine worked very smoothly, and could have been run at a higher speed and against a greater head without difficulty, and for any reasonable length of time.

DUTY TEST.

It should be stated that the coal used in this test was not of as good quality as contemplated in the contract, but was carefully hand-picked and gave very fair evaporative results. The duty has been figured for the entire run of 25.25 hours. The pressure, or head, against which the pumps worked was determined by means of the indicator, and also by means of gauges placed on the suction and force mains near the pumps, and again by the gauge on the engine, and the measured distance of this gauge above the level of the water in the well. The indicators were placed one on each water cylinder, and one on the suction and one on the discharge or force main. In the final report the duty will be figured for each set of observations, but in this report the duty is figured from the indicator diagrams taken from the water cylinders only.

Duration of tests, hours	25.25
Total number of revolutions	40,921
Average revolutions per minute	27.01
Average piston speed per minute, feet	135.05
Average pressure in water cylinders, pounds equivalent to a head of 136.44 feet.	59.0645
Total work done during the entire run, foot-pounds	3,802,999,500
Total coal fired to furnace, pounds	3,634.50
Added to make the fire as good as at beginning, pounds	140.00
Total coal charged up to test, pounds	3,774.50
Deducted as equivalent to the steam lost by leakage, which was caught and weighed, pounds	56.638
Net coal charged to the test, pounds	3,717.862

$$\text{Duty: Foot-pounds } \frac{3\,802,999,500}{3717.8620} \times 100 = 102,290,000$$

The duty as figured from gauge pressure involves an allowance for friction, which is included in the above result. This friction is variously estimated at from one pound per square inch up to several pounds, and the duty must of course vary accordingly. This will be fully considered in the final report.

The duty as above figured exceeds the requirements of the contract by 7.67 per cent.

I have the honor to remain,

Yours very truly,

M. E. COOLEY.

KALAMAZOO, MICH.

JULY, 1891.

GENERAL REPORT OF M. E. COOLEY,

PROFESSOR OF MECHANICAL ENGINEERING, UNIVERSITY OF MICHIGAN.

ANN ARBOR, Michigan, July 1, 1891.

To the Honorable the Common Council, Kalamazoo, Michigan—The Committee on Fire and Water:

GENTLEMEN—The preliminary report on the performance of the New Gaskill Pumping Engine at the Kalamazoo City Water Works, submitted February 28th, contained a copy of that portion of the contract bearing upon the capacity of the engine for domestic and fire service and the duty performance, together with a statement that the requirements of the contract had been fully complied with.

I have now the honor to submit a general report on this engine, containing in full the results obtained in the tests of January 6th, 7th, 8th and 9th, and in the special test of April 2d, made to determine the friction of the valves and the water passages of the pumps.

The test of April 2d and the report upon it have nothing to do with the requirements of the contract, and are introduced for the purpose only of making this report more complete, and for the knowledge afforded by them of the action of parts of pumping engines not usually considered.

Much gratification has been experienced in the opportunities afforded for securing this data, and it is believed that the information derived fully warrants the labor and care necessary to obtain it. Trusting that the report will be found acceptable, I have the honor to remain,

Yours very truly,

M. E. COOLEY.

[This is a duplicate of the engine illustrated and described on pages 98 and 99 of this book.—HOLLY MFG. CO.]

GENERAL REPORT ON THE NEW GASKILL PUMPING ENGINE, KALAMAZOO, MICHIGAN.

The Kalamazoo system of water works is known as the Direct System, the water being delivered by the pumps directly into the distributing mains of the city, no reservoir or stand-pipe being used.

The supply is taken from two wells, one on the north side and close to the pump house, the other 2,978 feet to the south and west. The well by the pump house is the older well, and has been in use 22 years. It is 20 feet in diameter inside the curb and about 25 feet deep. The new well was finished in 1890, and is 21 feet in diameter and about 25 feet deep. The water is brought from the new well to the old well by a siphon 24 inches in diameter, laid on a level between the wells. Valves are placed so that the pumps can draw directly from the new well if desired. Ordinarily the pumps take their supply from the old well.

The city is provided with four pumping engines, as follows: A Holly Quadruplex of two million gallons capacity, a Worthington Duplex of two and one-half million gallons capacity, and two Gaskill Compound Engines of three million gallons capacity each. The first two are now used only as a reserve.

The Gaskill Engines are of the Vertical Compound Condensing type, with crank and fly-wheels, and are duplicates, except that they are built right and left-handed. The older engine of the two stands on the north side of the pump house, and the new engine on the south side.

Each engine has one high and one low-pressure steam cylinder, and each cylinder operates by a direct connection a single-acting plunger pump.

The cross-head of each cylinder is connected by suitable links to a beam, the shaft of which is between the piston rods. The beam is further connected by means of a connecting-rod to a fly-wheel shaft carrying two fly-wheels, one at each end. The fly-wheels of the new engine are somewhat heavier than those of the old engine.

The steam cylinders are steam jacketed, both on sides and heads, and the sides are covered with wooden lagging. The valve gear is of the Corliss type, and may be set to give either a fixed or an automatic cut-off as desired. It is driven by a trussed rod attached to a pin in the end of the fly-wheel shaft.

The pump plungers are of cast iron, and packed by means of an external gland lined with bronze.

The valve chambers are in three parts; the lower part connects to the suction, the middle part to the pump cylinder, and the upper part to the discharge. The top plates of the lower and middle parts form the seats for the suction and discharge valves, respectively. The valves are of the type known as the "Troy Valve." The suction main at the pump is 16 inches in diameter and connects to a 20-inch main leading from the well and supplying both pumps. The pressure main is also 16 inches in diameter at the pumps.

Only one air chamber is used on the suction, serving for both the north and the south engines; but this is a large one and is situated on the 20-inch main just before it branches off to the pumps, and about 30 feet from the valve chambers of the south engine. The pressure main is provided with two air chambers, one of the usual form and size for these pumps above the valve chambers, the other about 25 feet distant. The latter air chamber is very large and is common to both engines.

The condenser, of the jet type, stands by the side of the engine and is connected to two single-acting trunk air-pumps. Two plunger feed-pumps are secured to the side of the air-pumps. The air-pumps and the feed-pumps are driven by a rock-sha which is an extension of the beam shaft.

The feed-water passes through a heater where it receives the heat of the jacke water discharged from an automatic steam-trap. The heater is a plain cylindrical sh standing in a vertical position containing a smaller cylindrical shell with a spa between them. The feed-water enters the bottom of the outer shell, passes up aroun the inner shell and leaves at the top, while the jacket-water enters the top of the inn shell, descends slowly and leaves at the bottom, discharging into the hot-well.

The boilers, two in number, are of the cylindrical tubular type, and are set in single battery. Both boilers are kept under steam, but only one is required to opera the pumps except in case of a fire, when both are used.

Previous to the test all gauges were compared with standards brought from t Mechanical Laboratory of the University of Michigan. These standards were verifi at the Laboratory both before and after the test. The indicators and gauges used the pumps and water mains, the thermometers, barometer and calorimeter were al brought from the University. The indicators used on the steam cylinders belonged the Water Works. All indicator springs were carefully tested at practically the tem peratures at which they were used in the test. The location of all instruments wa noted, and measurements taken to determine errors of position. All tabulated result have been corrected, both for errors of position and instrumental errors. The scal used to weigh the coal and ashes was tested with standard weights belonging to th city, both before and after the test.

The boiler room data was taken by the City Engineer, Mr. Frank C. Balch; th pump data by Professor Frank C. Wagner, of the University of Michigan, and th engine-room and well data by the writer. Mr. George H. Chandler, Superintendent of the Water Works, attended to the regulation of the machinery and assisted in operating the indicators on the steam cylinders.

The Holly Manufacturing Company was represented by Messrs. C. H. Parrott, J. W. Elliott and C. H. Anderson.

Mr. G. E. Balch, Assistant City Engineer, acted as time-keeper. The more important observations were taken on the stroke of the bell one minute after a warning stroke had been given, and the others as soon afterward as possible.

The capacity and duty tests began at 10.00 A. M., January 6th, and ended at 11.15 A. M., January 7th, the length of the test being 25.25 hours.

Observations were taken throughout the test as follows·

Every 15 Minutes.—Steam pressure at boiler, steam pressure at engine, vacuum in condenser, revolution counter, water pressure in pressure main by gauge on engine water pressure in pressure main by gauge near pumps, vacuum in suction main by gauge near pumps, temperature of feed-water at meter, temperature of feed-water a boiler, temperature of hot-well.

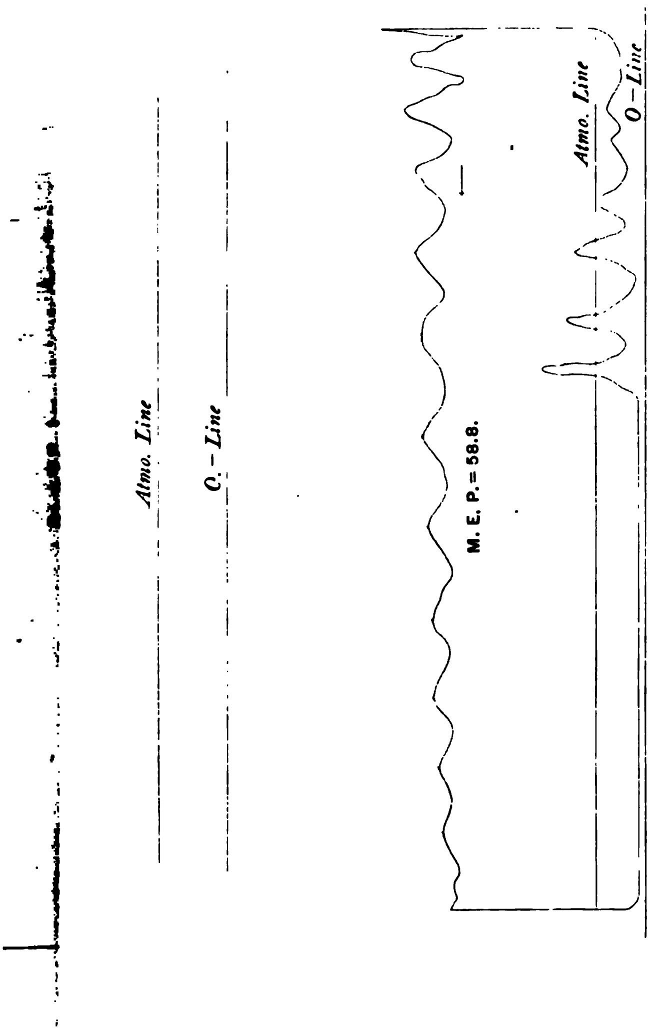
Every 30 Minutes.—Level of float in well, height of water in boiler, weight of coa delivered to fireman, revolutions counted.

Every Hour.—Temperature of engine-room, temperature of boiler-room, water meter reading.

Occasionally. — Barometer, temperature of external air, temperature of wate pumped, temperature of injection.

Indicator diagrams were taken every 30 minutes throughout the test, except between the hours of 11.00 P. M. and 6.00 A. M., when hourly cards were taken.

All of the water cards, and the hourly steam cards, have been worked up, and the results embraced in the tables of engine and pump performance.



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ASTOR, LENOX
TILDEN FOUNDATIONS

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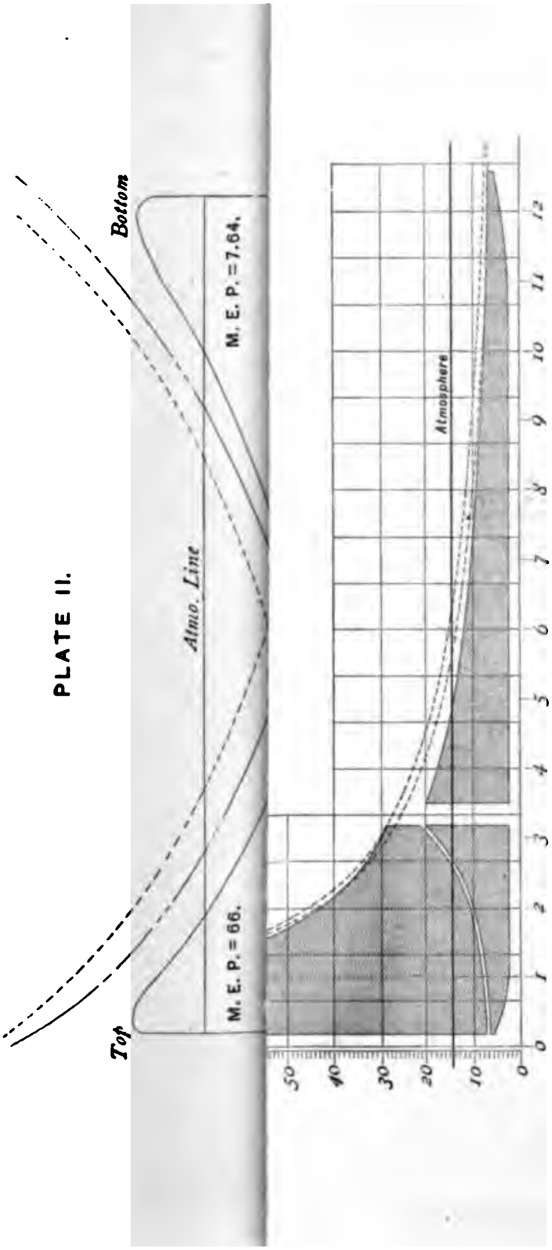
PLATE II.



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ASTOR, LENOX
TILDEN FOUNDATIONS

the hours of 11.00 P. M. and 6.00 A. M., when hourly cards were taken.

All of the water cards, and the hourly steam cards, have been worked up, and results embraced in the tables of engine and pump performance.

A full set of indicator diagrams is shown on Plate I, and a combined steam diagram on Plate II.

In the analysis of the test the natural order of boiler, engine, and pumps is followed, and for each is given a table of dimensions, a description of the test, and a table of performance.

The fire tests and the friction test are placed after the other tests, as they were made at later dates.

DIMENSIONS OF BOILER.

Type of boiler Horizontal tubular.	
Number of boilers	1.
Diameter of shell, inches	60.
Length of tubes, feet	16.
External diameter of tubes, inches	3.
Number of tubes	70.
Arrangement of tubes . Horizontal and vertical rows.	
Least distance between tubes and shell, inches	2.5
Width of vertical space in center, inches	3.
Height of crown of boiler above tubes, inches	19.5
Diameter of steam dome, inches	30.
Height of steam dome, inches	36.
Heating Surface:	
One-half the shell, square feet	125.664
One-half the heads, square feet	16.678
All the tubes, square feet	815.920
Total, square feet	958.262
Furnace:	
Width, feet	4.541
Depth, feet	2.833
Area, square feet	12.864
Draught area of tubes, square feet	2.957
Ratio of heating surface to grate area	74.491
Ratio of grate area to draught area of tubes	4.350
Chimney:	
Height, feet	110.
Length of side (square), feet	3.5
Draught area, square feet	12.25

BOILER TEST.

The boiler test was made simultaneously with the engine test. It was originally intended to begin the tests by hauling the fire and starting with weighed fuel; but as the fire had been newly cleaned and was heavy and in good condition, showing a bright ash-pit, it was decided to note the condition, and to bring it to the same condition at the end of the test. It was accordingly observed by three observers, and noted on the log.

The coal used was "Lehigh" lump, selected by the Holly Manufacturing Company, but was not considered to be of as good quality as contemplated in the contract, which specified "best coal." It was brought into the fire-room in a barrow, and was hand-picked, broken into a box on a platform scale, and weighed in charges of 70 pounds each. The scale was tested with U. S. standard weights both before and after the test and found correct.

Fifty charges were delivered to the fireman at half-hour intervals during the test; two extra charges were delivered, one after the test had been running sixteen and one-half hours, the other one hour before the end, making a total of fifty-two charges, or 3,640 pounds. From this is to be deducted 5.5 pounds weighed back, leaving a net weight of 3,634.5 pounds actually fired.

The firing was done by Mr. J. W. Elliott, of the Holly Manufacturing Company, in a very skillful manner.

Each 70-pound charge was divided into two parts, and fired alternately to the two sides of the furnace at intervals of 15 minutes. The draught being strong, the fire was carried deep on the grate, and it was found unnecessary to disturb it during the first sixteen and one-half hours; it was then shaken slightly and an extra charge added, gradually bringing the fire to its original level.

It became evident toward the end of the test that the fire was very dirty, as there were but few ashes in the ash-pit. Observing the depth to be slightly greater than at the beginning, it was again shaken about one-half hour before stopping, this time thoroughly. It fell away to such an extent that, instead of adding enough coal to restore the original level (which would have required considerable time to obtain conditions similar to those at the start), it was decided to haul the fire, weigh the coal and refuse, and compare directly with the conditions noted at the start. This was accordingly done. The ashes were first hauled from the ash-pit, and the fire then dumped and enough water used to barely quench it; the heat remaining was sufficient to evaporate the moisture and leave the refuse nearly dry. The clinkers were then separted, the remainder carefully screened, and the whole spread on the floor and left several hours, so that when finally weighed, but little moisture remained. The following table shows the weights of the several parts:

Ashes and clinkers from ash-pit, dry, pounds	124.375	
Coal from ash-pit, dry, pounds		19.875
Ashes from grate, slightly damp, pounds	72.000	
Clinkers from grate, pounds	98.000	
Coal from grate, pounds		260.000
Total coal from refuse, pounds		279.875
Total ashes and clinkers, pounds	294.375	294.375
Total from ash-pit and grate, pounds		574.250
Ashes in ash-pit before final shaking, estimated, pounds		60.000
Total from grate, pounds		514.250

The fire on the grate at the beginning of the test was estimated to be from 8 to 9 inches deep, or at an average, say, of 8.5 inches. The area of the grate was 12.86 square feet, and the volume of coal, therefore, 12.86 × 0.7 = 9.00 cubic feet, which, at 54 pounds per cubic foot, equals 486 pounds.

The actual weight of unconsumed coal on the grate at the beginning of the test was undoubtedly less than this, as the volatile matter and moisture, together amounting to about 6.16 per cent., must have disappeared. This would reduce the weight by $0.0616 \times 486 = 30$ pounds nearly, leaving 456 pounds as the weight on the grate at the beginning.

The difference between the computed weight of coal and ashes on the grate at the beginning, and the actual weight of coal found at the end of the test, appears as follows:

Coal and ashes at the beginning, computed, pounds	456.000
Coal at end, weighed, pounds	279.875
To be accounted for, pounds	176.125

It was accordingly agreed to add to the coal account two extra charges, or 140 pounds and to allow the balance, 36.125 pounds, to stand for ashes and other possible losses in the fire at the beginning. The total weight of coal charged to the test is, therefore, $3,634.5 + 140 = 3,774.5$ pounds.

The total weight of ashes found on hauling the fire was 294.375 pounds. This weight includes, of course, whatever ashes were in the fire at the beginning of the test, which should be deducted in order to find the true weight to be charged to the test.

Assuming 36.125 pounds as the weight to be deducted leaves the balance, $294.375 - 36.125 = 258.25$ pounds to be charged as ashes, which amount cannot be greatly in error.

The percentage of ashes and clinkers is then equal to $\frac{258.25}{3774.5} \times 100 = 6.844$, which, considering the care exercised in firing, may be regarded as a fair result.

The water was measured to the boiler by means of a brass Worthington water-meter situated midway between the heater and the boiler and connected in the usual way with a by-pass valve. The temperature of the water was taken by a thermometer screwed into the feed-pipe at the meter, and also by drawing water from a cock in the feed-pipe close to the boiler. Care was taken to prevent losses of water and steam to the other boilers. A blank flange was inserted in the pipe connecting the two adjacent boilers, the several valves in the feed-pipe leading to the old boilers were closed tightly, and the feed-pipe from the North Engine was disconnected. The steam valves of the several boilers were also closed, and the drips in the connecting pipes opened, one between the two adjacent boilers, one in the pipe leading from the old boilers, and one from the safety-valve case. The pipe supplying the steam heating system was disconnected and plugged. After the test had been running a short time the steam-pipe close to the North Engine was found to be warm, showing a leak; the drip-pipes from the cylinders and steam-jackets of this engine were then disconnected, and the leakages from them, as well as from the steam-pipes about the boilers, collected in buckets, weighed and entered on the log.

As an additional precaution the boiler not in use was kept under steam at nearly the same pressure as the boiler in use, and the height of water in the gauge was measured from time to time to note any change. Frequent and careful examinations failed to reveal any appreciable leakages other than from the drips as recorded. It was considered not advisable to separate the boiler in use from the other boilers and the North Engine by blank flanges, owing to the liability to be called on at any moment to respond to a fire-alarm, which might require more power than a single boiler and engine could supply.

The following tables show the losses by leakage, and the steam used in the calorimeter.

LEAKAGES ABOUT BOILERS.

	Time, Hours.	Number of Pounds.	Number of Pounds for Entire Test.
11.00 A. M. to 2.00 P. M	3.	30.5	
2.00 P. M. to 5.00 P. M	3.	27.0	
5.00 P. M. to 8.30 P. M	3.5	32.5	
8.30 P. M. to 11.30 P. M	3.	30.5	
11.30 P. M. to 2.00 A. M	2.5	20.5	
2.00 A. M. to 5.00 A. M	3.	31.5	
5.00 A. M. to 8.00 A. M	3.	43.5	
8.00 A. M. to 11.00 A. M	3.	31.0	
Totals	24.	246.0	
Average number of pounds per hour		10.25	
Total for 25.25 hours			258.812
SAFETY VALVE.			
5.00 P. M. to 6.00 P. M	1.	1.0	
Total for 25.25 hours			25.250

LEAKAGES ABOUT NORTH ENGINE.

	Time, Hours.	Number of Pounds.	Number of Pounds for Entire Test.
CYLINDER DRIPS.			
11.30 A. M. to 2.30 P. M	3.	15.00	
2.30 P. M. to 5.30 P. M	3.	14.00	
5.30 P. M. to 8.30 P. M	3.	13.50	
8.30 P. M. to 11.30 P. M	3.	14.25	
11.30 P. M. to 2.30 A. M	3.	13.00	
2.30 A. M. to 5.00 A. M	2.5	13.00	
5.00 A. M. to 8.15 A. M	3.25	15.00	
8.15 A. M. to 10.30 A. M	2.25	12.00	
Totals	23.	109.75	
Average number of pounds per hour		4.771	
Total for 25.25 hours			120.467
JACKET DRIPS.			
2.00 P. M. to 6.00 P. M	4.	15.50	
6.00 P. M. to 10.00 P. M	4.	16.50	
10.00 P. M. to 2.00 A. M	4.	15.00	
2.00 A. M. to 6.00 A. M	4.	13.75	
6.00 A. M. to 10.15 A. M	4.25	15.00	
Totals	20.25	75 75	
Average number of pounds per hour		3.74	
Total for 25.25 hours			94.452
CALORIMETER.			
11.40 A. M. to 12.40 P. M	1.	54.018	
6.00 P. M. to 7.00 P. M	1.	54.018	108.036
Total steam losses for 25.25 hours, pounds			607.019

As the evaporation was found to be high, a thermometer cup was inserted in the sheet-iron cover of the smoke flue at the rear end of the boiler, and the temperature taken with a Green thermometer during the latter part of the test; it was found to vary but little from 254° Fahr., which was considerably below the temperature of the steam. The temperature of the gases undoubtedly fell somewhat in passing from the front of the boiler through this flue about 20 feet to the thermometer, and it is fair to assume that the gases left the boiler at about the same temperature as the steam, so that there was no loss of heat up the chimney. This, together with the very large ratio of heating surface to grate area and the skillful firing, serve to explain what otherwise might appear to be an unusually high evaporation.

The water-meter was carefully tested by weighing the water discharged, with the following results:

No.	Pressure, Pounds.	Temperature, Fahr.	Number of Cubic Feet.	Total Weight, Pounds.	Weight per Cubic Foot, Pounds.
1	20.	101.	5.05	298.0	59.010
2	50 to 80	94.5	6.00	354.5	59.083
3	12.	104.0	6.00	353.5	58.916
4	40 to 80	101.5	6.00	354.5	59.083
Averages ...	....	100.25	....	....	59.023

The average temperature of the feed-water at the meter during the test was 119.37 degrees; and the weight per cubic foot at this temperature may be found by Rankine's formula, as follows:

$$D_{,} = \frac{2D_{\circ}}{\frac{T + 461}{500} + \frac{500}{T + 461}}$$ in which $D_{,}$ = weight of 1 cubic foot at a given temperature

T° Fah., and $D_{\circ}$ = weight of 1 cubic foot at 39°.1 Fah. = 62.425 pounds.

$$\text{At } 100^{\circ}.25,\ D_{,} = \frac{2 \times 62.425}{\frac{561.25}{500} + \frac{500}{561.25}} = \frac{124.85}{2.01337} = 62.01046 \text{ pounds.}$$

$$\text{At } 119^{\circ}.37,\ D_{,} = \frac{2 \times 62.425}{\frac{580.37}{500} + \frac{500}{580.37}} = \frac{124.85}{2.02226} = 61.738 \text{ pounds.}$$

Hence, 1 cubic foot by the meter at 119°.37 equals $\frac{61.738}{62.01046} \times 59.023 = 58.7638$ pounds.

PERFORMANCE OF BOILER.

Duration of test, hours	25.25
Fuel; anthracite coal; "Lehigh" lump:	
Total amount fired, pounds	3,634.50
Amount added to make fire as good as at beginning, pounds	140.00
Total amount charged to test, pounds	3,774.50
Ashes and clinkers; total amount charged to test, pounds	258.25
Combustible; total amount, pounds	3,516.25

Fuel; anthracite coal; "Lehigh" lump:	
Percentage of ashes and clinkers	6.844
Pressures:	
Steam pressure; average, pounds per square inch, gauge	79.025
Barometer; average, inches 29.754, pounds	14.611
Steam pressure; average, pounds per square inch, absolute	93.636
Temperatures; average, degrees Fahr.:	
Feed-water at meter	119.37
Feed-water at boiler	116.19
Flue-gases	254.30
Fire-room	79.70
External air	15.70
Water Level in Boiler; above bottom of gauge glass, inches:	
Height at beginning of test	7.8
Height at end of test	7.7
Average height during test	8.34
Water, as measured by meter:	
Meter reading at the beginning	607.00
Meter reading at the end	1295.60
Total number of cubic feet by meter	688.60
Weight of 1 cubic foot by meter	58.764
Total number of pounds fed to boiler	40,464.75
Quality of steam	dry.
Evaporation under actual conditions:	
Number of pounds of water per pound of coal	10.720
Number of pounds of water per pound of combustible	11.508
Factor of evaporation	1.1348
Evaporation from and at 212° Fahr.:	
Total number of pounds of water	45,920.36
Number of pounds of water per pound of coal	12.166
Number of pounds of water per pound of combustible	13.059
Rate of Combustion; per hour:	
Number of pounds of coal burned	149.485
Number of pounds of coal burned per sq. foot of grate area	11.620
Number of pounds of coal burned per sq. foot of heating surface	0.156
Rate of Evaporation under actual conditions; per hour:	
Number of pounds of water evaporated	1602.565
Number of pounds of water evaporated per sq. foot of grate area	124.577
Number of pounds of water evaporated per sq. foot of heating surface	1.672
Rate of Evaporation from and at 212 degrees; per hour:	
Number of pounds of water evaporated	1,818.627
Number of pounds water evaporated per sq. foot of grate area	141.374
Number of pounds of water evaporated per sq. foot of heating surface	1.898
Horse-powers developed on the basis of 34.5 pounds per H. P.	52.71
Disposition of Heat:	
Number of heat units per pound of coal, estimated	15000.
Number of heat units transmitted to water in boiler per pound of coal 12.166 × 966 =	11,752.

Disposition of Heat :

Temperature of furnace, degrees Fahr., estimated		2400.
Temperature of boiler room, degrees Fahr		80.
Temperature of furnace above temperature of boiler room, degrees Fah.		2,320.
Temperature of flue gases, the same as temperature of steam, degrees Fah.		323.
Temperature of the steam above temperature of boiler room, degrees Fah.		243.
Number of heat units passing to chimney, $\frac{243}{2320} \times 15000 =$		1,571.
Number of heat units accounted for		13,323.
Number of heat units lost by radiation and otherwise		1,677.
Percentage of heat transmitted to water in boiler	78.3	
Percentage of heat passing to chimney	10.5	
Percentage of heat lost by radiation and otherwise	11.2	
		100.

DIMENSIONS OF ENGINE.

(As taken from Contractor's Drawings.)

Stroke of pistons, nominal, inches	30.

High-Pressure Cylinder :

Stroke of piston, actual, inches	30.081
Diameter of piston, inches	18.
Diameter of piston rod (one), inches	2.625
Area of top of piston, square inches	254.47
Area of bottom of piston, square inches	249.06
Average area of both faces, square inches	251.76
Stroke displacement of top of piston, cubic inches	7,654.70
Stroke displacement of bottom of piston, cubic inches	7,491.90
Average stroke displacement, cubic inches	7,573.30
Volume of clearance and steam passages at one end of cylinder, cubic inches	411.60
Total volume of displacement and clearance, top of piston, cubic feet	4.668
Total volume of displacement and clearance, bottom of piston, cubic feet	4.579
Average volume of displacement and clearance, cubic feet	4.623
Ratio of clearance to stroke displacement of top of piston	.0536
Ratio of clearance to stroke displacement of bottom of piston	.0549
Ratio of clearance to average stroke displacement	.0543
Area of steam port, square inches	12.
Area of exhaust port, square inches	21.9

Low-Pressure Cylinder :

Stroke of piston, actual, inches	29.987
Diameter of piston, inches	36.
Diameter of piston-rods (two), inches	3.4375
Area of top of piston, square inches	1,017.878
Area of bottom of piston, square inches	999.317
Average area of both faces, square inches	1,008.597
Stroke displacement of top of piston, cubic inches	30,523.15
Stroke displacement of bottom of piston, cubic inches	29,966.50
Average stroke displacement, cubic inches	30,244.83

Low-Pressure Cylinder:

Volume of clearance and steam passages at one end of cylinder, cubic inches	775.5
Total volume of displacement and clearance, top of piston, cubic feet	18.113
Total volume of displacement and clearance, bottom of piston, cubic feet	17.790
Average volume of displacement and clearance, cubic feet	17.952
Ratio of clearance to stroke displacement of top of piston	0.0254
Ratio of clearance to stroke displacement of bottom of piston	0.0259
Ratio of clearance to average stroke displacement	0.0256
Area of steam port, square inches	21.9
Area of exhaust port, square inches	43.75

Condenser:

Exhaust pipe, diameter, inches	7.
Condenser, volume, cubic feet	8.97
Injection pipe, diameter, inches	4.

Air-Pumps:

Number	2.
Diameter, inches	12.
Stroke of piston, inches	14.5
Displacement of one air-pump piston, single stroke, cubic feet	0.942

Feed-Pumps:

Number	2.
Diameter, inches	3.
Stroke, inches	7.25
Displacement of one feed-pump, single stroke, cubic feet	0.0296

High-Pressure Connecting-Rod (one):

Length between centers, inches	26.
Diameter of journals, inches	4.
Length of journals, inches	4.5

Low-Pressure Connecting-Rods (two):

Length between centers, inches	26.
Diameter of journals, inches	4.
Length of journals, inches	2.75

Main Connecting-Rod:

Length between centers, inches	87.5625
Diameter of journal, beam end, inches	4.5
Length of journal, beam end, inches	5.25
Diameter of journal, crank end, inches	12.
Length of journal, crank end, inches	8.

Beam:

Distance between beam-center and connecting-rod centers, inches	23.25
Diameter of beam journals (two), inches	8.
Length of beam journals, inches	....

Fly-Wheels:

Number	2.
Diameter, feet	10.
Side of rim (square), inches	9.
Area of rim, square inches	81.

Fly-Wheels:

Weight of rim, pounds	7,353.	
Weight of six spokes, pounds	1,012.	
Weight of hub, pounds	1,260.	
Total weight of one fly-wheel, pounds		9,625.
Diameter of crank-shaft journals (two), inches		12.
Length of crank-shaft journals, inches		15.5

ENGINE TEST.

Under this head will be considered the action of that part of the machine devoted to the working of the steam.

The steam was taken from the top of the steam dome, and brought to the engine through about 10 feet of 6-inch pipe and 28 feet of 5-inch pipe, passing on its way three valves and eight right-angle turns. The pipe was thoroughly protected against losses by radiation, and the steam was delivered at the throttle free from moisture. The average steam pressure at the boiler was 79.025 and at the throttle 75.698 pounds per square inch by the gauge. The fall of pressure from the boiler to the throttle was then nearly 3.3 pounds; a slight additional fall took place on entering the cylinder, as shown by the indicator.

The valves of the high-pressure cylinder were set to cut-off at a little less than three-tenths of the stroke, slight changes being made during the night and in the morning as the consumption of water in the city changed.

The expansion of the steam was continuous from the high-pressure cylinder through the low-pressure, with a drop between them of about 1 pound.

The average final pressure at the end of the expansion in the low-pressure cylinder was 6.66 pounds absolute. The condenser pressure was 2.09 pounds. This fall in pressure, with the exception of about 1 pound at release, was gradual to the end of the return stroke, where it was sensibly equal to the condenser pressure.

The steam after being condensed was delivered by the air-pumps to the hot-well from which the boiler feed-pump took its supply, the balance going to waste.

The steam for the jackets was taken from the main steam-pipe near the engine, the pressure in the jackets being the same as at the throttle. The drip from the jackets was conducted to a Nason steam-trap close by the engine, from which it was discharged through the heater into the hot-well. The size of the discharge-pipe from the trap was ¾-inch and the length, 12 feet to the heater, and 19 feet from the heater to the hot-well. The feed pipe to the heater was 14.6 feet of 1¼-inch, and 6 feet of 2-inch pipe; and from the heater to the boiler, 24 feet of 2-inch pipe. The meter was placed 11 feet from the heater, and connected to the main feed-pipe with 7.6 feet of 1½-inch pipe. All pipes were covered with Standard Insulated Air Covering ⅞-inch thick.

The steam pressure in the main steam-pipe and jackets and the vacuum, were taken from the regular engine gauges on top of the engine. These gauges were provided with special connections for attaching standard gauges for comparison while the engine was in operation. Several comparisons were made during the test, and the differences noted on the log.

The revolutions were taken from the engine counter, and also by a different observer by counting, as a check upon the engine counter, the two results agreeing very closely.

Two Thompson Indicators were attached to the steam cylinders, one to each, and cards were taken every half-hour during the test, except between 11.00 P. M. and 6.00 A. M., when hourly cards were taken. Only the hourly cards have been worked up and used in this report, however.

The quality of the steam was determined by means of a Barrus Universal Calorimeter attached to the throttle-valve case. This instrument was in operation only twice: once between 11.40 A. M. and 12.40 P. M., and again between 6.00 and 7.00 P. M.

In both cases the steam was found to be perfectly dry.

The temperature of the discharge water from the condenser was taken from a thermometer screwed into the discharge pipe close to the hot-well; the temperature of the injection, being the same as that of the water pumped, was not separately observed.

With the exception of short intervals when elevators were in operation, or the railroad tank was being filled, the engine worked with smoothness and regularity. Most of the elevators were out of service during the test, and but little trouble was experienced from this cause. It should be noted, however, that when the elevators are in service, the variations in speed are at times somewhat excessive, and accompanied by violent slamming of the pump valves. It is unavoidable that these valves should slam when the number of revolutions exceeds a certain limit.

The acceleration of the engine due to a sudden fall of pressure in the water mains when a large valve, as an elevator valve for instance, is opened is to be expected, and is of course necessary, in order to restore the normal pressure under the increased flow.

The action of the fly-wheels is to retard this acceleration, so that a somewhat longer time is required for the engine to reach the necessary increased speed. Again with this increased speed, when the elevator valve is as quickly closed, the inertia of the fly-wheels tends to increase the pressure in the water mains beyond the normal during the time that is required to restore the original conditions of speed and pressure. It follows, therefore, that for sudden fluctuations of pressure in the pumps and water mains the fly-wheel is objectionable rather than beneficial.*

The principal function of the fly wheels is then to maintain a uniform speed of the engine under the variation of pressure due to expansion in the steam cylinder, and this apparently is well provided for.

A certain amount of fluctuation of pressure in the pumps and water mains is unavoidable in this class of pumps, and is due to the fact that both plungers come to rest at the same time. This flunctuation is reduced by the air chamber an amount depending upon its capacity. The fluctuations due to this cause observed during the test were not sufficient to be objectionable.

Although the jacket-water was not measured, it may with certain assumptions be figured with more or less accuracy. The arrangement of the feed-water heater was such that it is probable that the temperature of the jacket-water was reduced to the temperature of the feed-water entering the heater, which was the same as the temperature of the hot-well. The temperature of the jacket-water being known from the steam pressure, and the temperature and weight of the feed-water being known, the computation becomes very simple, and is as follows:

Temperature of jacket-water in degrees Fah., from the average steam pressure (90.31 pounds absolute)	320.280
Average temperature of hot-well	94.047
Fall in temperature of jacket-water	226.233
Average temperature of feed-water at meter	119.370
Average temperature of hot-well	94.047
Rise in temperature of feed-water	25.323
Total weight of feed-water per hour, pounds	1,602.11

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* The same objections apply to the direct-acting type of pumping engines, especially when a sudden fall of pressure in the water main occurs, as the pistons are not only instantly accelerated, causing sudden fluctuations of water pressure, but are likely to travel beyond their stroke and strike the cylinder heads, which cannot happen with a crank and fly-wheel engine.—HOLLY MFG. CO.

Number of heat units absorbed by feed-water per hour.......................40,570.23

Number of pounds of jacket-water, per hour = $\frac{40,570.23}{226.233}$ = ... 179.33

Percentage of jacket-water.. 11.36

The result is about what might be expected for this class of engines. The heater and all connecting pipes were thoroughly protected against loss by radiation. As the temperature of the feed-water was not far above the temperature of the basement it would lose only a small amount of heat. The steam-trap appeared to work properly, so that very little or no steam could have blown through it; and the pipe from the trap to the heater being small and short, it is not probable that it lost much heat.

The conclusion must follow that the result is at least not unreasonable.

PERFORMANCE OF ENGINE.

Duration of test, hours	25.25
Steam Used:	
Total number of pounds of water supplied to boiler	40,464.75
Total number of pounds of water due to leaks, caught and weighed	607.02
Total number of pounds of steam delivered to engine	39,857.73
Number of pounds of steam delivered to engine per hour	1,578.52
Quality of steam	Dry.
Coal Used:	
Total number of pounds charged to boiler	3,774.50
Total number of pounds equivalent to leakages $\frac{607.02}{10.72} =$	56.63
Total number of pounds charged to engine	3,717.87
Number of pounds charged to engine per hour	147.24
Combustible Used:	
Total number of pounds charged to boiler	3,516.25
Total number of pounds equivalent to leakages $\frac{607.02}{11.508} =$	52.75
Total number of pounds charged to engine	3,463.50
Number of pounds charged to engine per hour	137.09
Steam Pressures; average, pounds per square inch, gauge:	
At boiler	79.025
At engine	75.698
Barometer, inches 29.754; pounds	14.611
Steam Pressures; average, pounds per square inch, absolute:	
At boiler	93.636
At engine	90.309
Fall of pressure from boiler to engine, pounds	3.327
Vacuum in condenser, inches	25.54
Vacuum in condenser, pounds per square inch	12.52
Pressure in condenser, pounds per square inch	2.09

Temperatures; average, degrees Fahr.:	
Injection water	50.5
Discharge water, hot-well	94.05
Engine room	67.
External air	15.7
Revolutions:	
Counter reading at beginning	116,590.
Counter reading at end	157,511.
Total number of revolutions	40,921.
Average number of revolutions per hour	1,620.634
Average number of revolutions per minute by counter	27.01
Average number of revolutions per minute as counted	27.03
Piston Speeds; average, feet per minute:	
High pressure	135.414
Low pressure	134.991
Average of both pistons	135.202
Mean Effective Pressures; average of hourly cards, pounds per sq. inch:	
High pressure cylinder, top	46.796
High pressure cylinder, bottom	47.397
High pressure cylinder, average	47.096
Low pressure cylinder, top	7.178
Low pressure cylinder, bottom	8.200
Low pressure cylinder, average	7.689
Total Effective Pressures on Pistons; pounds:	
High pressure piston, top	11,908.177
High pressure piston, bottom	11,804.700
High pressure piston, average	11,856.438
Low pressure piston, top	7,306.340
Low pressure piston, bottom	8,194.400
Low pressure piston, average	7,750.370
Top of high pressure and bottom of low pressure pistons	19,214.517
Bottom of high pressure and top of low pressure pistons	19,999.100
Indicated Horse-Powers:	
High pressure cylinder, top	24.429
High pressure cylinder, bottom	24.216
High pressure cylinder, total	48.645
Low pressure cylinder, top	14.942
Low pressure cylinder, bottom	16.758
ssure cylinder, total	31.700
th engines	80.345
n of steam per indicated horse-power per hour	19.647
f coal per indicated horse-power per hour	1.8326
ombustible per indicated horse-power per hour	1.7062

STEAM ACCOUNTED FOR BY THE INDICATOR; average of 5 cards:

(See table following.)

Number of pounds of steam delivered to engine per hour	1,578.52
Number of pounds of steam delivered to engine per stroke	0.487
Pounds per Stroke; average of both strokes:	
High pressure at cut-off (0.33 stroke)	0.2930
High pressure at release	0.3364
Low pressure at beginning (0.04 stroke)	0.2780
Low pressure at release	0.3229
Pounds per Hour; average of both strokes:	
High pressure at cut-off (0.33 stroke)	949.53
High pressure at release	1,090.17
Low pressure at beginning (0.04 stroke)	900.91
Low pressure at release	1,046.42
Percentage of Steam Accounted For; average of both strokes:	
High pressure at cut-off (0.33 stroke)	60.17
High pressure at release	69.08
Low pressure at beginning (0.04 stroke	57.08
Low pressure at release	66.31
*Number of pounds of steam delivered to jackets per hour, computed	179.4
Number of pounds of steam delivered to cylinders per hour	1,399.064
Number of pounds of steam delivered to cylinders per stroke	0.4316
Percentage of steam delivered to jackets	11.36
Percentage of Steam Accounted For in Cylinders; average of both strokes.	
High pressure at cut-off (0.33 stroke)	67.88
High pressure at release	77.94
Low pressure at beginning (0.04 stroke)	64.41
Low pressure at release	74.81
Percentage of Steam in Cylinders for Adiabatic Expansion (theoretical):	
High pressure at cut-off (0.33 stroke)	67.88
High pressure at release	65.30
Low pressure at beginning (0.04 stroke)	64.78
Low pressure at release	62.98

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* See computation.

PRESSURES OF STEAM IN THE CYLINDERS, AS SHOWN BY SELECTED INDICATOR DIAGRAMS.

Number.	Time.	Point of Cut-off.		Boiler Pressure.	Initial Pressure.		Pressure at 0.33 Stroke.		Final Pressure, H. P. Cylinder.		Back Pressure at 0.04 of Return Stroke, H. P. Cylinder.		Back Pressure at End of Return Stroke, H. P. Cylinder.		Pressure at .04 Stroke, L. P. Cylinder.		Final Pressure, L. P. Cylinder.		Back Pressure, Beginning of Return Stroke, L. P. Cylinder.		Back Pressure, End of Return Stroke, L. P. Cylinder.	
		Top.	Bottom.		Top.	Bottom.	Top.	Bottom.	Top.	Bottom.	Top.	Bottom.	Top.	Bottom.	Top.	Bottom.	Top.	Bottom.	Top.	Bottom.	Top.	Bottom.
1	12.50 P. M.	.279	.290	78.	78.	78.	62.3	65.0	15.5	17.5	4.8	7.3	*7.3	*6.6	4.0	4.4	*8.1	*7.4	*9.0	*8.4	*12.2	*12.0
2	5.45 P. M.	.278	.298	78.	80.7	80.7	60.8	68.2	15.0	17.5	4.7	7.2	*7.8	*6.7	3.4	4.4	*8.2	*7.0	*8.9	*8.2	*12.1	*11.9
3	10.45 P. M.	.281	.299	77.	79.2	78.7	62.0	65.6	15.2	17.5	5.2	7.6	*7.0	*6.2	3.0	4.2	*8.3	*7.2	*9.1	*8.2	*12.2	*11.9
4	4.45 A. M.	.271	.291	79.	80.5	80.5	62.1	66.3	14.9	18.0	5.3	8.0	*7.5	*6.5	3.4	4.3	*8.1	*7.1	*9.0	*8.9	*12.1	*12.1
5	10.00 A. M.	.270	.283	77.	80.0	79.8	60.0	64.6	14.5	17.3	5.0	7.5	*7.4	*6.5	2.85	3.8	*8.2	*7.3	*9.2	*8.6	*12.2	*12.0
Average	....	.276	.292	77.8	79.7	79.5	61.44	65.94	15.02	17.56	5.0	7.52	*7.4	*6.5	3.33	4.22	*8.18	*7.2	*9.04	*8.47	*12.16	*11.98
Corrected	....	....	....	79.4	76.12	75.93	58.68	62.98	14.34	16.77	4.77	7.18	*7.07	*6.21	3.44	4.86	*8.45	*7.44	*9.34	*8.75	*12.56	*12.38
Absolute Pressures	....	....	....	94.01	90.73	90.54	73.29	77.59	28.95	31.38	19.38	21.79	7.54	8.40	18.05	18.97	6.16	7.17	5.27	5.86	2.05	2.23

* Negative pressures.

DIMENSIONS OF PUMPS.

Stroke of plungers, nominal, inches	30.
Diameter of plungers, nominal, inches	20.
Diameter of plunger rods (two to each), inches	3.
High-Pressure Plunger:	
Stroke, actual, inches	30.081
Diameter, actual, inches	19.998
Area, square inches	314.1056
Stroke displacement, cubic inches	9,448.608
Stroke displacement, cubic feet	5.4679
Stroke displacement, gallons	40.903
Low-Pressure Plunger:	
Stroke, actual, inches	29.987
Diameter, actual, inches	20.013
Area, square inches	314.5653
Stroke displacement, cubic inches	9,432.872
Stroke displacement, cubic feet	5.4588
Stroke displacement, gallons	40.835
Total displacement per revolution, cubic inches	18,881.48
Total displacement per revolution, cubic feet	10.9267
Total displacement per revolution, gallons	81.738
Valves:	
Style, brass with rubber face	"Troy."
Number of suction valves in each pump	91.
Number of discharge valves in each pump	91.
Total number of valves in both pumps	364.
Diameter of rubber valve discs, inches	1.75
Thickness of rubber valve discs, inches	0.5625
Weight of one valve, pounds	0.5412
Diameter of clear opening in seat, inches	1.3125
Lift of valve, inches	0.3125
Area of rubber valve disc, square inches	2.405
Area of clear opening in one valve seat, square inches	1.353
Area of cylindrical opening under one valve face when open, square inches,	1.286
Total area through valve seats in one set, square inches	123.12
Total area through valve seats in one set, square feet	0.8550
Total area under valve face when open in one set, square inches	117.26
Total area under valve face when open in one set, square feet	0.8143
Ratio of greater valve area to plunger area	0.391
Ratio of lesser valve area to plunger area	0.341
Weight of one valve per square inch of opening in seat, pounds	0.400
Weight of one valve per square inch of area of bottom of disc, pounds	0.225
Suction Main:	
Length, approximately, feet	75.
Diameter for 14.5 feet, inches	16.
Diameter for 60.5 feet, inches	20.
Area of the 16-inch pipe, square inches	201.06

Suction Main:

Area of the 16-inch pipe, square feet	1.396
Area of the 20-inch pipe, square inches	314.16
Area of the 20-inch pipe, square feet	2.182

Force Main:

Diameter, inches	16.
Area square inches	201.06
Area square feet	1.396

PUMP TEST.

The water was taken from the well on the north side of the pump house and brought to the pumps through about 60.5 feet of 20-inch pipe and 14.5 feet of 16-inch pipe, the smaller pipe being next to the pumps. The 20-inch main contained three long sweep quarter-turn bends, and was connected to the 16-inch pipe at right angles to it by a tee. The water was delivered by the pumps through a 16 inch pipe directly into the city mains, no reservoir or stand-pipe being used. The excess of water over the regular consumption was wasted through a hydrant, a man being stationed at the hydrant to regulate its discharge as the consumption in the city varied. In this way a practically uniform pressure and speed were maintained throughout the test. The elevator service about the city was generally discontinued. At times, however, when elevators were surreptitiously used, and on two occasions of about five minutes each when the railroad tank was being filled, the pumps speeded up considerably.

The pump valves were not examined at the time of the test, but it was apparent from their action that they were in good order. They had been examined a few days before the test and found to be perfect. The priming valves were carefully examined before the test, and at several times during the test, and were found to be tight. The plunger packing was in good order, and the pumps worked without undue friction.

The contract piston speed of 135 feet per minute was exceeded slightly during the test, the actual speed being 135.202 feet. As the contract stipulates that the capacity of the pumps shall be determined by the plunger displacement, there was no occasion to measure the slip; nor would it have been possible except by elaborate and expensive preparations. In several instances where the slip has been measured, it has been found not to exceed 2.5 per cent. for this class of pumps.

The capacity per day at 135 feet piston speed was found to be 3,174,466 U. S. standard gallons, which exceeds the contract requirement of 3,000,000 gallons by 174,466 gallons, or 5.815 per cent. Assuming a maximum probable slip of 2.5 per cent., there remains still a very safe margin of 3.315 per cent. excess.

The head against which the pumps worked was determined in several ways, as follows: By indicator placed on the water cylinders, by indicators placed on the suction and pressure mains, by gauges placed on the suction and pressure mains, and by the regular gauge on the engine and a float in the well. These several methods were used in order to secure as much data as possible covering the action of the pumps, valves, and connecting pipes and passages.

Tabor Indicators with 60-pound springs and a special suction spring were used on the water cylinders, one at the top of each, and connected by short ¾-inch nipples and reducing elbows. Thompson Indicators with 40 and 10-pound springs were used on the mains; the indicator on the pressure main was attached to a short 4-inch pipe on the bottom of the main by ¾-inch connections, and the indicator on the suction main to the top of the pipe by ½-inch connections. Two very fine spring gauges were placed no the mains, one by the side of each indicator, and connected to the mains by the same

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pipes. The pressure gauge was graduated in one-pound divisions, and the vacuum gauge in ½-inch divisions. The arrangement of these instruments is shown on Plate III.

The vertical distances of the pistons of the indicators on the water cylinders and pressure main, and of the center of the pressure gauge, from the center of pump plunger displacement, were carefully determined as follows: For the indicators on water cylinders, 19.75 inches; for the indicator on the pressure main, 9 inches; and for the pressure gauge, 13.25 inches. The corresponding pressures are 0.713, 0.325, and 0.478 pounds respectively, which are additive for pressures, and the first subtractive for suctions.

The correction for the indicator and gauge on the suction main is the distance from the top of the pipe to the center of plunger displacement, instead of from the centers of the instruments, and is equal to 12.25 inches, or 0.442 pounds, which is additive. This may at first sight appear to be incorrect, but when it is considered that a partial vacuum exists in the connections of these instruments, the tendency is for the water to draw out of them, leaving them empty or nearly so.

In a series of experiments performed in the Mechanical Laboratory at the University of Michigan, in which the connections of a gauge on the suction of a pump were made of glass tubing, no water was found to stand above the suction pipe; the water fluctuated with the motion of the pump, shooting up and disappearing with every stroke. It will also be apparent that unless the air is allowed to escape from the connections of pressure gauges, an error may arise in correcting for their position. The pressure gauge connections were allowed to leak a trifle at the cock in this test, and no error of that kind existed.

A datum was established by the city engineer, Mr. Frank C. Balch, at a convenient point on the platform in the well 18 feet below the center of the gauge on the engine. The vertical distance between this datum and the water level was measured by a copper float placed in a pipe to prevent swaying with the water currents, and connected by a cord over a pulley to a weighted index. A Chesterman tape-line graduated in tenths of a foot was tacked to a support behind the index, so that at the beginning of the test the zero of the tape was at the water level. The distance to the water was then read directly on the tape. The temperature of the water pumped was taken by drawing water from the pressure main above the valve chambers.

The stroke of the pump plungers was measured by securing a smooth pine board between the plunger and cross-head in a vertical position, and tracing a series of lines upon its surface with a finely-pointed hard pencil held firmly on the frame. The board was then removed and the lines measured by two observers with a 36-inch Starrett rule graduated in one-hundredths of an inch. This rule was afterwards compared with a 36-inch Brown & Sharpe standard, and found to agree at 30 inches. The measurements of the different lines were as follows:

H. P. PLUNGER.			L. P. PLUNGER.		
No. 1	30.085	inches.	No. 1	30.000	inches.
2	30.080	"	2	29.980	"
3	30.080	"	3	29.990	"
4	30.085	"	4	29.980	"
5	30.080	"	5	29.990	"
6	30.080	"	6	29.990	"
7	30.080	"	7	29.980	"
8	30.080	"			
9	30.080	"			
Average	30.081	"	Average	29.987	"

The diameter of the plungers was measured both with calipers and by means of a very thin steel tape. The caliper points were carefully beveled to an edge on the same side and rounded off, and the measurements taken on the Starrett rule.

The Starrett rule and the tape were found on comparison with a Brown & Sharpe Standard not to agree at the points of measurement. The Brown & Sharpe rule was assumed to be correct. The corrected measurements were as follows:

Calipered Measurements :

Diameter of H. P. plunger, inches	19.986
Diameter of L. P. plunger, inches	19.996

Tape Measurements :

Circumference of H. P. plunger, inches	62.840
Circumference of L. P. plunger, inches	62.886
Diameter of H. P. plunger by tape, inches	20.00255
Diameter of L. P. plunger by tape, inches	20.01718
Average thickness of tape, inches	0.00428
Diameter of H. P. plunger as corrected, inches	19.9983
Diameter of L. P. plunger as corrected, inches	20.0129

As it requires much skill to make accurate measurements of large bodies with calipers, especially when the bodies are wet, it is safer to depend on the tape. The thickness of the tape should, however, be allowed for.

A series of measurements was made after the test in the Mechanical Laboratory at the University of Michigan on a carefully turned and polished 16-inch pulley, with a view to testing the relative accuracy of the two methods of measurement. The thickness of the tape was measured at intervals of 1 foot with a micrometer caliper as follows :

Distance (feet).	*Thickness (inches).*
0	0.0044
1	0.0042
2	0.0044
3	0.0042
4	0.0042
5	0.0044
Average	0.00428

The pulley was carefully calipered on five different diameters by three individuals with almost identical results, as follows :

Number.	*Diameter (inches).*
1	16.046
2	16.044
3	16.052
4	16.052
5	16.046
Average	16.048

The circumference was then measured with the steel tape and corrected by the standard rule, as follows:

Tape measurement, inches	50.425
Corrected measurement, inches	50.428
Diameter by tape measurement, inches	16.05170
Average thickness of tape, inches	0.00428
Corrected diameter by tape, inches	16.04742
Diameter as calipered, inches	16.04800
Difference, inches	0.00058

The surface of the pulley was then wetted and the tape measurement repeated, but the presence of water on the surface did not appear to make any difference in the reading.

The head pumped against was found by four different methods and the duty figured separately for each, the results being given in the table of pump performance. The duty of this pumping engine, as stated and understood by the contract, is the number of foot-pounds of work done in the water cylinders by 100 pounds of coal as burned under the boiler. In calculating the duty of a pumping engine from pressures found by gauges, or by gauge and measurement, it is customary to add a certain amount for friction of valves and water passages. More or less uncertainty necessarily attends this assumption, as the friction varies greatly in different classes of pumps. The pressure found by indicators attached to the water cylinders includes the friction of valves and water passages, and is the most correct measure of the work done in the water cylinders.

The duty performance of this engine as found by half-hourly indicator cards from the pump cylinders is then the proper duty, and is that which was given in the preliminary report. Its corrected value is 102,289,741 foot-pounds, which is in excess of the contract requirement of 95,000,000 by 7,289,741 foot-pounds, or 7.673 per cent.

The other duties which are given in the table do not include any allowance for friction whatever, and are a correct measure of the net or useful work done by the engine. There is a very fair agreement in the duties as found by the indicators and by the gauges on the pressure and suction mains. The duty found by the gauge on engine and float should be slightly less than either of the others, as it does not include the frictional resistance of the suction main, or of a sharp turn in the pressure main. The gauge pipe was connected to the pressure main about 20 feet from the engine, the gauge itself being above the steam cylinders, so that the connecting pipe was about 30 feet long. It is possible, and perhaps probable, that a slight constant error was made in reading this gauge on account of its position. As the observer stood on the engine platform the gauge was about four feet to the front and left, and eight feet above the platform. The position of the pointer was such as to give an apparent reading higher than the true reading.

PERFORMANCE OF PUMPS.

Duration of test, hours	25.25
Steam Used:	
Total number of pounds charged to pumps	39,857.73
Number of pounds charged to pumps per hour	1,578.52

Coal Used:		
Total number of pounds charged to pumps		3,717.87
Number of pounds charged to pumps per hour		147.24
Combustible Used:		
Total number of pounds charged to pumps		3,463.50
Number of pounds charged to pumps per hour		137.09
Water Pressures by Gauge on Engine and Float; average per square inch:		
By gauge, pounds		39.774
Distance of float below datum, feet	18.547	
Distance of datum below gauge on engine, feet	18.000	
Distance of float below gauge on engine, feet	36.547	
Pressure equivalent to distance of float below gauge on engine, pounds.		15.834
Total head pumped against, pounds		55.608
Total head pumped against, feet		128.343
Reduced to center of plunger displacement:		
Force main, pounds		48.842
Suction main, pounds		6.766
Total head pumped against, pounds		55.608
Total head pumped against, feet		128.343
Water Pressures, by Gauges, in Mains near Pumps; average per square inch reduced to center of plunger displacement:		
Force main, pounds		47.390
Suction main, inches—13.55; pounds		6.643
Total head pumped against, pounds		54.033
Total head pumped against, feet		124.708
Water Pressures, by Indicators, in Mains near Pumps; average, pounds per square inch reduced to center of plunger displacement:		
Force main, H. P. stroke		46.456
Force main, L. P. stroke		46.107
Force main, average of both strokes		46.281
Suction main, H. P. stroke		7.440
Suction main, L. P. stroke		7.786
Suction main, average of both strokes		7.613
Total head pumped against from averages, pounds		53.894
Total head pumped against from averages, feet		124.387
Water Pressures, by Indicators, in Pump Cylinders; average, pounds per square inch reduced to center of plunger displacement:		
H. P. discharge		50.535
H. P. suction		8.312
Total head pumped against in H. P. cylinder		58.847
L. P. discharge		50.124
L. P. suction		9.158
Total head pumped against in L. P. cylinder		59.282

Water Pressures, by Indicators, in Pump Cylinders; average, pounds per square inch reduced to center of plunger displacement:

Total head pumped against, average of both cylinders	59.064
Total head pumped against, average of both cylinders, feet	136.32
Temperature of water pumped, average, degrees Fah	50.5
Weight of one gallon at 50.5 Fah., pounds	8.342

Revolutions:

Total number	40,921.
Average number per hour	1,620.634
Average number per minute	27.01

Plunger Speeds; average, feet per minute:

H. P. pump	135.414
L. P. pump	134.991
Average of both plungers	135.202

Velocity of Flow; feet per second:

In suction and discharge mains	3.603
Through valve seats	5.882
Through cylindrical openings under valve face when open	6.176

Water pumped; in cubic feet:

Total number by H. P. pump	223,753.8
Total number by L. P. pump	223,381.0
Total number by both pumps	447,134.8
Number per hour by H. P. pump	8,861.54
Number per hour by L. P. pump	8,846.77
Number per hour by both pumps	17,708.31

Water Pumped; in U. S. standard gallons:

Total number by H. P. pump	1,673,794.0
Total number by L. P. pump	1,671,006.0
Total number by both pumps	3,344,800.0
Number per hour by H. P. pump	66,288.90
Number per hour by L. P. pump	66,178.46
Number per hour by both pumps	132,467.36

Capacity per day; in U. S. standard gallons:

Number at plunger speed during the test, 135.202 feet per minute	3,179,216.0
Number at contract plunger speed of 135 feet per minute	3,174,466.0
Contract requirement	3,000,000.0
Number in excess of contract requirement	174,466.0
Percentage of excess over contract requirement	5.815

Total Effective Pressures against Plunger by Indicators on Water Cylinders; pounds:

H. P. plunger	18,484.17
L. P. plunger	18,648.07
Average of both plungers	18,566.12
H. P. discharge and L. P. suction	18,754.45
L. P. discharge and H. P. suction	18,377.79
Average of both plungers	18,566.12

WORK; foot-pounds:	
By gauge on engine and float, total	3,580,925,628.0
By gauges on mains near pumps, total	3,479,505,760.0
By indicators on mains near pumps, total	3,470,548,800.0
By Indicators on Pump Cylinders:	
Total by H. P. pump	1,896,082,969.0
Total by L. P. pump	1,906,916,667.0
Total by both pumps	3,802,999,636.0
Number per minute by H. P. pump	1,251,539.4
Number per minute by L. P. pump	1,258,690.6
Number per minute by both pumps	2,510,230.0
Indicated horse-power of H. P. pump	37.925
Indicated horse-power of L. P. pump	38.142
Indicated horse-power of both pumps	76.067
Economic Results:	
* Duty by gauge on engine and float, no allowance being made for friction of water passages	96,316,587.0
Duty by gauges on mains near pumps, no allowance being made for friction of water passages	93,588,660.0
Duty by indicators on mains near pumps, no allowance being made for friction of water passages	93,347,744.0
Duty by indicators on pump cylinders	102,289,741.0
Contract requirement	95,000,000.0
Number in excess of contract requirement	7,289,741.0
Percentage of excess over contract requirement	7.673
Number of pounds of steam per net horse-power per hour	20.752
Number of pounds of coal per net horse-power per hour	1.9356
Number of pounds of combustible per net horse-power per hour	1.8022
Friction of Engine:	
Indicated horse-powers in steam cylinders	80.345
Indicated horse-powers in pump cylinders	76.067
Frictional horse-power	4.278
Percentage of friction	5.324

FIRE TEST.

The tests for fire service were made on January 8th, in the evening, so as to interfere as little as possible with the elevator service. No length of time was specified for the test to continue. The test was understood to be for the purpose of determining whether the pumps were capable of withstanding easily and safely the severe stresses incident to fire service. For this purpose it was necessary to continue the test only long enough to become satisfied that the required work was being performed, and that there were no signs of weakness in any part of the machinery.

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* Duty = number of foot pounds per 100 lbs. coal.

The contract required that the engine pump at the rate of 3,000,000 gallons per day against a head of 125 pounds per square inch, at a piston speed of 135 feet per minute.

Four short tests were made, each under different conditions. The first was a preliminary test, and the last was made at the request of the Superintendent, Mr. Chandler, to determine the capacity of the engine when working compound.

The counter, steam pressure at engine, water pressure at engine, and vacuum, were taken every minute; and the float in the well, at the beginning, in the middle, and at the end of the test.

In fire service the cut-off is disconnected and steam allowed to follow full stroke in the high-pressure cylinder, the speed of the engine being controlled by the throttle. The high-pressure cylinder is provided with a 1¼-inch by-pass connecting the two ends of the cylinder so as to admit live steam directly into the low-pressure cylinder. This enables the engineer to increase the power of the engine so as to pump against such pressure as may be required in fire service.

In the first test, the by-pass valve was about one-quarter open, perhaps less, and the discharge pipe to the city was wide open. The average steam pressure was 73.03, and the pressure against the pumps was 107.65, pounds per square inch, the piston speed being 158.335 feet per minute. It soon became evident that without a considerable increase of speed, the water pressure could not be raised to 125 pounds per square inch unless the discharge pipe was throttled. The gate just outside the pump house was accordingly closed partially.

In the second test the by-pass valve was wide open, the throttle-valve about one-quarter open, and the discharge main partially closed. The average steam pressure was 66.7, and the pressure against the pumps was 128.66, pounds per square inch, the piston speed being 154 feet per minute. At the high piston speeds of this and the first test the pumps worked smoothly, but with a marked slamming of the pump valves.

In the third test the discharge main was closed still more, so as to reduce the piston speed and at the same time increase the pressure. The by-pass valve was wide open, and the throttle-valve about two-thirds open until near the end, when it was opened wide as the steam-pressure fell. The steam pressure, and the pressure pumped against, were maintained almost constant. The average steam pressure was 72.8, and the pressure against the pumps 132.05, pounds per square inch, the piston speed being 139.75 feet per minute. The engine worked very smoothly, and it was evident that the test could have been continued for any reasonable length of time, and even at a higher pressure against the pumps.

In the fourth test the bye-pass valve was closed and the engine was operated as a plain compound engine, with throttle wide open, the steam following full stroke in the high pressure cylinder. The average steam pressure was 73.3, and the pressure against the pumps 124.89, pounds per square inch, the piston speed being 130.1 feet per minute.

This result is very gratifying, as it shows that without using the by-pass valve the engine is capable of fulfilling the contract requirements very nearly, even at a reduced steam pressure. It is more than probable that with a steam pressure of 75 pounds per square inch, or slightly above, the full pressure of 125 pounds per square inch against the pumps and the full piston speed of 135 feet per minute could be realized. The tabulated results of the four tests are as follows:

RESULTS OF FIRE TESTS MADE JANUARY 8, 1891.

Conditions of Test.	1	2	3	4
	By-pass open slightly. Discharge wide open.	By-pass wide open. Discharge partly closed. Reduced steam pressure. Throttle ¼ open.	By-pass wide open. Discharge still more closed. Throttle ⅝ open.	**By-pass closed. Discharge partly closed. Throttle wide open.**
Test began, P. M.	8.47	9.25	9.45	**10.22**
Test ended	8.53	9.35	10.05	**10.32**
Duration of test, minutes	6.	10.	20.	**10.**
Total number of revolutions	190.	308.	559.	**261.**
Number of revolutions per min.	31.667	30.8	27.95	**26.1**
Piston speed per minute, feet	158.33	154.	139.75	**130.5**
Steam pressure at engine, pounds per square inch	73.03	66.7	72.8	**73.3**
Vacuum in condenser, inches	23.66	19.75	20.05	**24.20**
Water level in well by float, feet.	18.3	18.93	18.675	**18.25**
Elevation of gauge on engine, feet	18.0	18.00	18.000	**18.00**
Water level in well below gauge on engine, feet	36.3	36.93	36.675	**36.25**
Water level in well below gauge on engine, pounds	15.714	16.00	15.87	**15.71**
Water pressure in pressure main, pounds	87.714	108.50	111.95	**104.95**
Friction of pump valves and passages, pounds	4.23	4.23	4.23	**4.23**
Total pressure pumped against, pounds	107.66	128.73	132.05	**124.89**
Contract requirement, pounds	125.	125.	125.	**125.**
Percentage in excess of, or less than, contract requirement	(—) 13.87	2.984	5.64	**(—) 0.088**
Capacity per day, U. S. gallons	3,722,958.	3,621,143.	3,286,070.	**3,068,566.**
Contract requirement, gallons	3,000,000.	3,000,000.	3,000,000.	**3,000,000.**
Percentage in excess of contract requirement	24.10	20.70	9.53	**2.28**

FRICTION OF PUMP VALVES AND WATER PASSAGES.

The friction of the valves and the water passages of a pump is not often obtained in a pumping engine test, as it is not usually convenient to secure the requisite data. In this test, however, suitable arrangements had been made, so that it was very convenient to attach the necessary instruments and to secure the data.

The importance of knowing what the friction is in a pump test becomes apparent in considering the error that may creep into the computation for duty, especially when the head against which the pump works is small. It is usual to find this head by means of gauges on the pressure and suction mains, or by a gauge on the pressure main and by measuring from the center of this gauge to the level of the water from which the supply is taken. To the head thus obtained a certain quantity is added for friction of pump valves and water passages, giving a total head which is used for the duty computation. The amount added for friction is generally one pound, unless specified or otherwise understood. As a matter of fact the friction is usually much greater than one pound, reaching even thirteen pounds and more in certain recorded instances.

Had the total head against the pumps in this test been found by adding one pound to the head as given by the gauges on the pressure and suction mains, the duty would have been 95,313,580 foot-pounds, instead of 102,289,741 as actually found in the pump cylinders. The error amounts to 6.82 per cent.

It is not to be inferred that the friction of these pumps is unusually high. On the contrary, it is what might reasonably be expected, considering that the piston speed is above the normal. It is probable that, as pumps go, the friction will be found greater instead of less than the friction in these pumps, in many cases two or three times as great.

In numerous cases on record where the utmost care has been taken in reading the gauges and in measuring the distance to the water level of the supply, the results may be many per cent. in error, owing to the questionable practice of adding "one pound for friction of valves and water passages."

In determining the friction of pumps it is necessary, especially where the speed varies as in a direct pumping system, to take observations simultaneously. In the duty test the cards and readings about the pumps were taken by a single observer, and a minute or two necessarily elapsed between the first and last readings, during which both the speed of the pumps and the pressure in the mains might have varied considerably. The data thus obtained was not sufficient to admit of separating the friction of the suction valves from that of the discharge valves. The friction, therefore, as given by the duty test, is the average total friction for each pump, and the average total friction for both.

An opportunity presenting itself, new series of cards and observations were taken on April 2d, with special reference to determining the friction of the suction and discharge valves separately. In this test the cards were taken simultaneously from the pump cylinders and mains close to the pumps. Special care was taken to keep the indicators free from dirt and grit. Cards were never taken until the speed of the engine was uniform, and then revolutions were counted with a stop-watch reading to fifths of a second. Usually five whole revolutions were counted, and the time recorded. If there was the least evidence of change of speed the observations were repeated.

Two Tabor Indicators were used on the water cylinders, a 60-pound spring in one, and a 10-pound special suction spring in the other. Two Thompson Indicators were used on the mains, a 40-pound spring in one, and a 10-pound special suction spring in the other. The suction cards were taken exactly together, the pencil being applied at the beginning and removed at the end of the stroke; and the pressure cards in the same way. Three sets of cards were taken, and then the springs were interchanged on the water cylinders, and three more sets taken. This was done for the ordinary working speed of the pumps, 18 to 19 revolutions per minute, and also for a speed of from 26 to 27 revolutions per minute.

The indicator springs were very carefully tested at the temperatures at which they were used, at the Mechanical Laboratory of the University of Michigan. A special weighing device was made use of for that purpose, with which it was possible to secure very great accuracy. The diagrams were worked up by means of three different planimeters which had also been tested for accuracy. No pains have been spared to obtain the most reliable results. That they are so concordant warrants the hope that they will be found of more than passing interest, and of value to the engineering profession.

In securing the data for the tables which follow, the time and the revolutions were taken by Mr. G. E. Balch, the indicator cards from the pump cylinders, by Mr. M. C. Taft, and those from the mains, by the writer.

DIMENSIONS OF PUMPS AND WATER PASSAGES.

High-Pressure Plunger:

Stroke, actual, inches	30.081
Diameter, actual, inches	19.998
Stroke displacement, cubic feet	5.4679

Low-Pressure Plunger:

Stroke, actual, inches	29.987
Diameter, actual, inches	20.013
Stroke displacement, cubic feet	5.4588

Valves:

Style, brass with rubber face	"Troy."
Number in each set	91.
Diameter of rubber valve disc, inches	1.75
Thickness of rubber valve disc, inches	0.5625
Weight of one valve, pounds	0.5412
Diameter of clear opening in seat, inches	1.3125
Lift of valve, inches	0.3125
Area of rubber valve disc, square inches	2.405
Area of clear opening in one valve seat, square inches	1.353
Area of cylindrical opening under one valve face when open, sq. inches.	1.286
Total area through valve seats in one set, square inches	123.12
Total area through valve seats in one set, square feet	0.855
Total area under valve face, when open, in one set, square inches	117.26
Total area under valve face, when open, in one set, square feet	0.8143
Ratio of greater valve area to plunger area	0.391
Ratio of lesser valve area to plunger area	0.341
Weight of one valve per square inch of opening in seat, pounds	0.400
Weight of one valve per sq. inch of area of bottom of rubber disc, lbs.	0.225

[illegible]:

[illegible] of suction main at pump, inches	16.0
[illegible] of pressure main at pump, inches	16.0
[illegible] 16-inch main, square inches	201.06
[illegible] 16-inch main, square feet	1.396

[illegible] FROM DUTY TEST OF JANUARY 6 AND 7, 1891.

[illegible]

[illegible] against, by indicator on pump cylinders, pounds.		58.847
[illegible] main during H. P. discharge, by indicator, pounds.	46.456	
[illegible] main during H. P. suction, by indicators, pounds.	7.440	
[illegible] against, by indicators on mains, pounds		53.896
[illegible] pump, pounds		4.951

Low-Pressure Pump:

Total head pumped against, by indicator on pump cylinders, pounds.		59.282
Pressure in force main during L. P. discharge, by indicator, pounds.	46.107	
Pressure in suction main during L. P. suction, by indicator, pounds.	7.786	
Total head pumped against, by indicators on mains, pounds........		53.893
Total friction of L. P. pump, pounds............................		5.389
Average friction of both pumps, pounds.........................		5.17

By Gauges:

Average total head pumped against, by indicators on pump cylinders, pounds		59.064
Average pressure in force main, by gauge, pounds	47.590	
Average pressure in suction main, by gauge, pounds	6.643	
Average total head pumped against, by gauges, pounds		54.033
Average total friction of pumps, pounds..........................		5.031

Velocity of Flow; feet per second :

In suction and discharge mains..................................	3.603
Through valve seats..	5.882
Through passage under valve face, when open....................	6.176

FRICTION TEST OF APRIL 2, 1891.

LOW SPEED SERIES.

Cards taken { Suction main and H. P. suction / Pressure main and L. P. discharge } together.

Number of Observation.	Number Counted.	REVOLUTIONS. Suction. Time, Seconds.	Suction. Revolutions per Minute.	Discharge. Time, Seconds.	Discharge. Revolutions per Minute.
1	5	15.4	19.48	16.0	18.75
2	5	15.2	19.74	15.4	19.48
3	6	19.6	18.37	....	.....
	5	....	.....	16.4	18.29
Averages..			19.20		18 84

Cards taken { Suction main and L. P. suction / Pressure main and H. P. discharge } together.

4	5	15.8	18.99	17.8	16.85
5	5	16.2	18.52	16.2	18.52
6	5	15.8	18.99	16.6	18.07
7	5	15.2	19.74	16.8	17.86
8	5	15.4	19.48	15.4	19.48
Averages..			19.14		18.16

LOW SPEED SERIES.

MEAN EFFECTIVE PRESSURES BELOW ATMOSPHERE FROM SUCTION CARDS.

Number of Observation.	H. P. Pump Cylinder.		L. P. Pump Cylinder.		Suction Main.		Differences.	
	Observed.	Corrected.	Observed.	Corrected.	Observed.	Corrected.	H. P.	L. P.
1	6.89	6.74			3.77	3.69	3.05	
2	6.49	6.34			3.53	3.45	2.89	
3	6.71	6.56			3.82	3.74	2.82	
Averages		6.547				3.627	2.92	
Correction for position		0.713				0.442	1.155	
Corrected averages		5.834				4.069	1.765	
4			7.37	7.22	4.53	4.45		2.77
5			6.91	6.76	4.51	4.43		2.33
6			7.04	6.89	4.12	4.04		2.85
7			7.05	6.90	4.30	4.22		2.68
8			6.83	6.68	4.27	4.19		2.49
Averages				6.89		4.266		2.624
Correction for position				0.713		0.442		1.155
Corrected averages				6.177		4.708		1.469

MEAN EFFECTIVE PRESSURES ABOVE ATMOSPHERE FROM DISCHARGE CARDS.

Number of Observation.	H. P. Pump Cylinder.		L. P. Pump Cylinder.		Discharge Main.		Differences.	
	Observed.	Corrected.	Observed.	Corrected.	Observed	Corrected.	H. P.	L. P.
1			48.30	49.30	47.76	46.88		2.42
2			49.68	50.68	50.88	50.00		0.68
3			48.54	49.54	49.64	48.76		0.78
Averages				49.84		48.547		1.293
Correction for position				0.713		0.325		0.388
Corrected averages				50.553		48.872		1.681
4	48.54	49.54			48.28	47.40	2.14	
5	47.46	48.46			47.44	46.56	1.90	
6	46.38	47.38			46.00	45.12	2.26	
7	51.30	52.30			51.20	50.32	1.98	
8	50.64	51.64			50.56	49.68	1.96	
Averages		49.864				47.816	2.048	
Correction for position		0.713				0.325	0.388	
Corrected averages		50.577				48.141	2.436	

SUMMARY OF RESULTS.

High-Pressure Suction:	
Average number of revolutions per minute	19.20
Average M. E. P. below atmosphere in pump cylinder, pounds	5.834
Average M. E. P. below atmosphere in suction main, pounds	4.069
Average friction, pounds	1.765
Low-Pressure Suction:	
Average number of revolutions per minute	19.14
Average M. E. P. below atmosphere in pump cylinder, pounds	6.177
Average M. E. P. below atmosphere in suction main, pounds	4.708
Average friction, pounds	1.469
Average friction of suction valves and passages, both pumps, pounds	1.617
High-Pressure Discharge:	
Average number of revolutions per minute	18.156
Average M. E. P. above atmosphere in pump cylinder, pounds	50.557
Average M. E. P. above atmosphere in force main, pounds	48.141
Average friction, pounds	2.436
Low-Pressure Discharge:	
Average number of revolutions per minute	18.84
Average M. E. P. above atmosphere in pump cylinder, pounds	50.553
Average M. E. P. above atmosphere in force main, pounds	48.872
Average friction, pounds	1.681
Average friction of discharge valves and passages, both pumps, pounds	2.058
Total Average Friction:	
High pressure pump, pounds	4.201
Low pressure pump, pounds	3.150
Both pumps, pounds	3.675

HIGH SPEED SERIES.

Cards taken {Suction main and H. P. suction / Pressure main and L. P. discharge} together.

NUMBER OF OBSERVATION.	REVOLUTIONS.				
	NUMBER COUNTED.	SUCTION.		DISCHARGE.	
		Time, Seconds.	Revolutions per Minute.	Time, Seconds.	Revolutions per Minute.
1	5	11.2	26.78	11.4	26.32
2	5	10.8	27.78	11.4	26.32
3	5	11.4	26.32	11.2	26.78
Averages..	....	....	26.96	....	26.47

HIGH SPEED SERIES.

Cards taken { Suction main and L. P. suction / Pressure main and H. P. discharge } together.

NUMBER OF OBSERVATION.	REVOLUTIONS.				
	NUMBER COUNTED.	SUCTION.		DISCHARGE.	
		Time, Seconds.	Revolutions per Minute.	Time, Seconds.	Revolutions per Minute.
4	5	11.0	27.27	11.0	27.27
5	5	11.6	25.86	11.2	26.78
6	5	11.4	26.32	11.2	26.78
Averages..			26.48		26.94

MEAN EFFECTIVE PRESSURES BELOW ATMOSPHERE FROM SUCTION CARDS.

NUMBER OF OBSERVATION.	H. P. PUMP CYLINDER.		L. P. PUMP CYLINDER.		SUCTION MAIN.		DIFFERENCES.	
	Observed.	Corrected.	Observed.	Corrected.	Observed.	Corrected.	H. P.	L. P.
1	9.25	9.10			5.75	5.67	3.43	
2	8.96	8.81			6.11	6 03	2.78	
3	7.95	7.80			5.09	5.01	2.79	
Averages.....		8.57				5.57	3.00	
Correction for position....		0.713				0.442	1.155	
Corrected averages....		7.857				6.012	1.845	
4			8.25	8.10	5.27	5.19		2.91
5			8.06	7.91	5.19	5.11		2.80
6			8.33	8.18	5.45	5.37		2.81
Averages.....				8.063		5.223		2.840
Correction for position....				0.713		0.442		1.155
Corrected averages....				7.350		5.665		1.685

HIGH SPEED SERIES.

MEAN EFFECTIVE PRESSURES ABOVE ATMOSPHERE FROM DISCHARGE CARDS.

Number of Observation.	H. P. Pump Cylinder.		L. P. Pump Cylinder.		Discharge Main.		Differences.	
	Observed.	Corrected.	Observed.	Corrected.	Observed.	Corrected.	H. P.	L. P.
1	....	....	47.58	48.58	47.88	47.00	....	1.58
2	....	....	47.70	48.70	47.52	46.64	....	2.06
3	....	....	47.46	48.46	48.08	47.20	....	1.26
Averages.....	....	....	....	48.58	....	46.947	....	1.633
Correction for position....	....	....	....	0.713	....	0.325	....	0.388
Corrected averages....	....	....	....	49.293	....	47.272	....	2.021
4	49.44	50.44	....	....	49.12	48.24	2.20	....
5	47.88	48.88	....	....	47.48	46.60	2.28	....
6	50.04	51.04	....	....	49.28	48.40	2.64	....
Averages.....	....	50.12	....	....	....	47.747	2.373	....
Correction for position....	....	0.713	....	....	....	0.325	0.388	....
Corrected averages....	....	50.833	....	....	....	48.072	2.761	....

SUMMARY OF RESULTS.

High-Pressure Suction:

Average number of revolutions per minute	26.96
Average M. E. P. below atmosphere in pump cylinder, pounds	7.857
Average M. E. P. below atmosphere in suction main, pounds	6.012
Average friction, pounds	1.845

Low-Pressure Suction:

Average number of revolutions per minute	26.48
Average M. E. P. below atmosphere in pump cylinder, pounds	7.350
Average M. E. P. below atmosphere in suction main, pounds	5.665
Average friction, pounds	1.685
Average friction of suction valves and passages, both pumps, pounds	1.765

High-Pressure Discharge:

Average number of revolutions per minute	26.94
Average M. E. P. above atmosphere in pump cylinder, pounds	50.833
Average M. E. P. above atmosphere in force main, pounds	48.072
Average friction, pounds	2.761

Low-Pressure Discharge:

Average number of revolutions per minute	26.47
Average M. E. P. above atmosphere in pump cylinder, pounds	49.293
Average M. E. P. above atmosphere in force main, pounds	47.272
Average friction, pounds	2.021
Average friction of discharge valves and passages, both pumps, pounds	2.391

Total Average Friction:

High pressure pump, pounds	4.606
Low pressure pump, pounds	3.706
Both pumps, pounds	4.156

CONCLUSION.

It remains to be stated that the results of the several tests show the contract requirements to have been exceeded in every case.

In closing I desire to acknowledge the courtesy of the Committee on Fire and Water, Messrs. Stearns, Gleason and Cellem, of the Superintendent, Mr. Geo. H. Chandler and his corps of able assistants, of the City Engineer, Mr. F. C. Balch and his assistant, and of the representatives of the Holly Manufacturing Company, Messrs. Parrott, Anderson and Elliott.

For the careful working of the large number of indicator cards, the comparison of instruments, and for numerous independent and check computations I am indebted to Professor F. C. Wagner. I have the honor to remain,

Yours very truly,

M. E. COOLEY.

COLUMBUS, OHIO.

JULY, 1891.

REPORT OF ROBT. B. COLLIER, M. E., AND S. P. BUSH, M. E.

COLUMBUS, O., July 23, 1891.

To the Honorable Board of Public Works, Columbus, O.:

GENTLEMEN—The undersigned, appointed by you to make capacity and duty tests of the new 7,500,000-gallon Gaskill Triple Expansion Pumping Engines at the East End Water Station, respectfully submit the following report:

The contract made with the Holly Manufacturing Company requires them to furnish two pumping engines of the triple-expansion type, capable of delivering 7,500,000 gallons of water per 24 hours each, against a head of 185 feet, exclusive of friction in pumps and connecting pipes, at a piston speed of 135 feet per minute; and to perform a duty of 120,000,000 foot-pounds based on a consumption of 100 lbs. of coal and an evaporation of 10 lbs. of water per pound of coal from and at the temperature of the feed water; and to pump against a pressure of 120 lbs. per square inch for fire service.

METHOD OF CONDUCTING TEST.

The test was made of the south engine, or No. 318, on a 12-hour run, beginning at 9.30 A. M., and concluding at 9.30 P. M., July 21st. All readings were taken at intervals of one-half hour from all the attachments, except the indicator diagrams, which were taken every hour.

The gauges used were as follows:

One steam gauge on each of the two boilers used, and one at the high pressure cylinder of the engine. One water gauge connected with the main. One vacuum gauge showing the vacuum in the condenser. A revolution counter, and one steam engine indicator attached to each cylinder.

The water levels from the surface of the water in the well to the center of the water pressure gauge were ascertained by means of a steel tape and an engineer's level.

While no requirements were specified as to the performance of the boilers, the coal consumed (Hocking Valley) was properly weighed, so that the evaporation of the boilers might be known when using this kind of coal.

The water fed to the boiler was run into a wooden tank having a capacity of about 600 lbs., and was then run into another wooden tank having about the same capacity, from which it was fed to the boiler. The first tank was set on platform scales which had been properly adjusted. It should be here stated that none of the condensation from the steam jackets was used for feed water, the feed being taken entirely from the main.

PRINCIPAL DIMENSIONS OF ENGINES.

Steam Cylinders:

High Pressure	24 in. Diameter.	48 in. Stroke.
Intermediate	38 in. Diameter.	48 in. Stroke.
Low Pressure	61 in. Diameter.	48 in. Stroke.

Pumps:

Three (3), Single-acting Plunger, outside packed, 26 in. Diameter × 48 in. Stroke.

WATER LEVELS IN BOILERS.

Inches of Water in Glass Tubes, 9.30 A. M.:

Boilers	No. 1.	No. 2.
Levels	2½ in.	3 in.

Inches of Water in Glass Tubes, 9.30 P.M.:

Boilers	No. 1.	No. 2.
Levels	2½ in.	3 in.

PERFORMANCE OF BOILERS.

Average steam pressure, pounds	124.46
Total coal weighed, pounds	7,935.
Total ash weighed, pounds	1,000.
Net combustible, pounds	6,935.
Water evaporated at temperature of feed, pounds	52,204.
Water evaporated per pound of coal, pounds	6.57
Water evaporated per pound of combustible, pounds	7.52
Temperature of feed water, degrees (constant)	53.

ENGINE ROOM READINGS.

	Initial.	*Final.*
Steam gauge	122.	124.
Water gauge	65.	65.
Vacuum gauge	26.	25.75

RESULTS FOR CAPACITY.

Mean head pumped against, feet	213.40
Counter, initial	355,986.
Counter, final	367,627.
Revolutions in 12 hours	11,641.
Revolutions per minute	16.16
Piston speed per minute, feet	129.28
Displacement per revolution, gallons	330.96
Number of gallons delivered in 12 hours	3,852,705.
Capacity per 24 hours, gallons	**7,705,410.**

From the above it may be observed that during the trial the engine made an average of 16.16 revolutions per minute, or a piston speed of 129.28 feet, and exceeded the contractors' guarantee by 205,000 gallons, or 2.73 %.

RESULTS FOR DUTY.

Engine counter, initial	355,986.
Engine counter, final	367,627.
Revolutions in 12 hours	11,641.
Revolutions per minute	16.16
Piston speed per minute	129.28
Mean head by water pressure gauge, feet	151.87
Mean distance from center of water pressure gauge to level of water in well, feet	61.53
Total head exclusive of friction, feet	213.40
Temperature of water in well, degrees F	52.
Weight of one gallon of water, pounds	8.33
Displacement per revolution, gallons	330.96
Water at one stroke, pounds	2,756.89
Total weight of water weighed to boiler, pounds	52,204.
Duty (foot-pounds per 1,000 pounds of steam)	131,205,000.

From the above it may be observed that the actual duty is 11,205,000 foot-pounds in excess of the guarantee, or 9.33 %. This does not take any account of friction in the supply pipe and connections, which is estimated to equal a head of one-half foot.

SUMMARY OF INDICATOR DIAGRAMS.

Cylinders.	*M. E. P.*	*H. P.*
High Pressure	51.67	91.04
Intermediate	20.50	95.92
Low pressure	11.06	126.97

Water per horse-power per hour, pounds	13.85

During the test one fire alarm came in at 12 o'clock, at which time the water pressure increased from 66 pounds to 69 pounds. At 12.07 P. M., the fire was declared out, and the water pressure resumed its normal condition.

Careful examinations of the engine were made at intervals, and it was found to work smoothly, and without heating.

Respectfully submitted,

ROBT. B. COLLIER,
S. P. BUSH.

PORTLAND, OREGON.

AUGUST, 1891.

REPORT OF ISAAC W. SMITH, SUPT. CITY WATER WORKS.

OFFICE OF THE CITY WATER WORKS,
PORTLAND, August 10, 1891.

The Water Committee, City of Portland, Oregon:

GENTLEMEN—I submit the result of the duty test of the pump constructed by the Holly Manufacturing Company, guaranteed with 80 lbs. steam pressure and a piston speed not exceeding 100 feet per minute, to pump at the rate of 2,500,000 gallons in 24 hours, against a water pressure, including suction lift, of 127 feet, with a duty of 80,000,000 foot pounds from the evaporation of 1,000 pounds of water.

The water fed to boilers was weighed without deduction for leakage or steam used for other purposes than pumping.

The water pumped was measured from the displacement of the plungers indicated by the revolution counters, without deduction for slip.

With due allowance for leakage and slip the duty would probably have been increased from 2 to 4 per cent.

The trial was from 9 A. M. to 9 P. M., August 8th, during which time careful records were taken at intervals of 15 minutes of counter, steam and vacuum gauges, and weight of water fed to boilers.

Indicator diagrams for steam and water were also taken, and in this and all other respects, the performance of the machinery was uniformly satisfactory during the whole time of the test.

SUMMARY OF OBSERVATIONS.

Duration of test	12	hours.
Average steam pressure on engine gauge	82	pounds.
Vacuum (observed)	27¼	inches.
Plungers, 15 inch diameter, 22 inch stroke, with 2½ inch rods.		
A—Displacement per revolution, 66.38 gallons =	553.14	pounds.
B—Revolutions (30.48 per minute) in 12 hours	21,946	
C—Water pumped (A × B) = 1,457,000 gallons =	12,139,210	pounds.
Average pressure on gauge 51.23 pounds =	118.34	feet.
Height from surface in well to gauge	17.33	feet.
D—Water pressure including suction lift	135.67	feet.
E—Foot pounds = C × D =	1,646,926,620	
F—Weight of water fed to boiler	15,706	pounds.
G—Duty = $\frac{E}{F} \times 1{,}000$ =	104,860,000	pounds.
Duty by contract	80,000,000	pounds.
Excess over contract duty	24,860,000	
Cords of wood per 1,000,000 gallons 135.67 lift	$\frac{58}{100}$	

The water was fed to boiler *cold*, otherwise the cords per million gallons would have been less.

Respectfully submitted,

ISAAC W. SMITH,
Superintendent City Water Works.

DULUTH, MINNESOTA.

AUGUST, 1891.

REPORT OF TRACY LYON, M. E., AND H. E. STEVENS, C. E.

ST. PAUL, Minn., Aug. 14, 1891.

S. M. CAREY, ESQ., *President Duluth Gas and Water Co., St. Paul, Minn.:*

DEAR SIR—In accordance with your instructions, the following report on the condition, capacity and duty of the 5,000,000 gallon Gaskill Pumping Engine, in operation at the pumping station of the Duluth Gas and Water Co., and on the fulfillment of the contract under which the engine was furnished by the Holly Manufacturing Co. the manufacturers, as far as its construction is concerned, is respectfully submitted.

The engine was first carefully examined and measurements made of all its parts.

The engine was found to fulfill in arrangement, in the number of parts, in the connections and valves, fittings and in dimensions, the requirements of the drawings and specifications of April 7th, 1890, upon which the contract above referred to was based.

The "Holly Hydrostatic Automatic Pressure Regulator," to be used in connection with the engine when pumping directly into the mains, without the intervention of the reservoir, is in place and apparently complete, but it has not been connected with the mains, or tested for such direct service; but has been used only for convenience in regulating the cut-off in the high-pressure steam cylinders.

The only further exception to the fulfillment of the contract is in the lack of paint, the machinery having received only the priming coat applied at the shop. The engine should be painted without delay, inasmuch as the studs of the water-cylinders are beginning to show evidence of wasting from the action of the water on the unprotected iron.

The design, material and workmanship of the engine are admirable, the only apparent defect being in the roughness of the castings. This roughness and lack of finish of the castings do not, however, detract in any way from the efficiency of the engine.

The engine, which was first turned over December 27th, 1890, began its regular pumping service January 1, 1891, running during the month of January, however, only from six to nine hours per day, part of the service being taken care of by the 3,000,000 gallon Worthington Pump in the station. From the first of February (with the exception of three days in that month) to date, the engine has performed the entire service of pumping necessary for the City of Duluth, having delivered, on an average, about 3,000,000 gallons of water per twenty-four hours. During this time the number of hours which the engine was run daily, ranged from 9 to 24, the maximum continuous run being 96 hours.

The actual material used for repairs during the seven months that the engine has been in operation, has consisted only of a wrought iron band used to stop the leakage from the feed-water heater until a new cover could arrive, and a steel pin in the bottom of one of the air-pump plungers which had to be replaced, the cost of this material not exceeding $3.00.

The present condition of the engine is excellent. There does not seem to be a crack or flaw in any part, and the engine as a whole runs smoothly and without undue heating or noise. This excellent condition is due in a very large degree to the efficiency and unremitting attention of the engineer in charge of the station, Mr. James Stewart.

TEST.

On August 6, 1891, the subscribers met in Duluth Mr. Charles Anderson, who represented the Holly Manufacturing Co., of Lockport, N. Y., the contractors, and arrangements were made for a capacity and duty test of the engine. Previous to this date an engineer of the Holly Mfg. Co. had been adjusting and preparing the engine for this test.

Following is an extract from the specifications stating the conditions and requirements of the test :

"CAPACITY.—The machinery above described will be guaranteed to be capable of lifting five million U. S. gallons of water per twenty-four hours, pump-plunger displacement, against a head of 285 feet, at a piston speed not to exceed 120 feet per minute."

"DUTY.—The engine will be guaranteed to perform a test-duty on a run of twenty-four hours, of 105,000,000 foot-pounds, when pumping at its rated capacity, against a static head of 285 feet, and the frictional head in suction, pump and passages, and 3,760 feet 20-inch cast iron pipe, based upon an evaporation by the boilers of ten pounds of water per pound of coal, from and at the temperature of the feed-water, with a steam-pressure of 95 pounds per square inch at the throttle valve, when running."

It was agreed to take half-hourly observations at the counter, steam, water and vacuum gauges on the engine, and the distance from the center of the water-gauge to the surface of the water in the pump-well, which was read by a float-guage; this latter measurement varying from time to time by some seven inches, according to the level of the lake.

The feed-water for the boilers was taken directly from the force main into a wooden tank capable of holding about 1,000 pounds, on platform scales, weighed, and run thence into a second and lower tank, which was connected directly with the boiler pump, it being necessary to use only one of the four boiler pumps connected with the engine. After leaving the pump, on its way to the boilers the water passed through a tubular heater, into which was trapped the condensation from the live steam in the jackets before passing into the hot-well. Ordinarily the feed-water is taken from the hot-well, but during the trial all communication with the hot-well was cut off. The second tank was full at the beginning of the test, and was filled to the same extent at the close of the test, with water weighed and charged to the engine, the hour and minute at which this second tank was filled each time being entered on the log.

Inasmuch as the specifications required the duty to be based upon an evaporation by the boilers of ten pounds of water per pound of coal, or 1,000 pounds of steam instead of 100 pounds of coal, it was not necessary for the test of the engine to determine the efficiency of the boilers. It seemed advisable, however, under the circumstances, to do this, and arrangements were therefore made to weigh the coal burned, and take the temperature of the feed-water at the boiler. Two boilers only, Nos. 3 and 4 of the battery of four in the station, were used, as they were amply sufficient to supply the engine with steam. The fires were cleaned half an hour before the test began, and three times during the test, the last time half an hour before the close. The height of the water in the gauge glasses, and the condition of the fires were made as nearly as possible the same at the end of the test as at the beginning.

Coal used was a good quality of Mansfield (Ohio) slack, and was brought on the floor in 200-pound charges as needed, the time brought in being noted on the log. It was fired as evenly as possible, each furnace being fired lightly at intervals of six to eight minutes.

Half-hourly observations were taken of the steam gauges on the boilers and the temperature of the feed-water. This latter observation was taken by means of a thermometer in a cup filled with oil set in the water pipe immediately before entering the boilers.

The test was begun at 3 P. M. on Thursday, August 6th, and continued until 3 P. M., Friday, August 7th, an uninterrupted period of 24 hours.

The management of the engine during the test was exclusively under the direction of the engineers of the Holly Manufacturing Co.

The boilers were fired by the regular firemen.

No allowances were made for the live steam used in the jackets, or for the work done by the boiler feed pump connected with the engine.

GENERAL DESCRIPTION OF ENGINE.

Diameter high pressure cylinders	24 inches.
Diameter low pressure cylinders	42 inches.
Diameter pump plungers	19½ inches.
Stroke of both steam pistons and pump plungers	36 inches.
Diameter H. P. piston rods	3 inches.
Diameter L. P. piston rods	3½ inches.
Diameter pump plunger rods	4 inches.
Diameter fly-wheel	13 feet 6 inches.
Number of air pumps, single acting	4
Number of boiler pumps	4

Steam cylinders jacketed throughout, jackets fed with live steam from boilers.
Jet condenser.
Tubular heater, heated by jacket condensation.

Discharge of pumps per revolution	182.212 gallons.
Percentage of slip in pump (assumed)	2.
Net discharge of pumps per revolution	178.568 gallons.

The regular service of the engine consists in pumping from a well connected with an in-take pipe running out into the lake, through 3,760 feet 20-inch cast iron pipe into a reservoir of some 5,000,000 gallons capacity, the bottom of which is 260 feet above Lake Superior. During the test the height of water in the reservoir was kept as nearly uniform as possible by the use of a blow-off valve on the 20-inch force main, the average height being 18 feet, thus giving a total static head of 278 feet.

GENERAL DESCRIPTION OF BOILERS.

Number of boilers (plain tubular)	2
Diameter of shell	60 inches.
Length of tubes	16 feet 6 inches.
Number of tubes	78
Diameter of tubes	3 inches.
Diameter of domes	30 inches.
Height of domes	30 inches.
Fire grates, each	5 × 4 feet.
Area grate surface	20 square feet.
Total heating surface, each	1,065 square feet.
Ratio of heating surface to grate surface	53.25 to 1.

Boiler setting—Standard full front.
Grates—Double ribbed "Tupper."
Boilers manufactured by Geo. F. Blake Manufacturing Co., 1883.

MEANS AND TOTALS OF OBSERVED DATA.

Duration of test, hours	24
Number of observations	49
Number of revolutions of engine in 24 hours	29,373
Mean number of revolutions of engine per minute	20.398
Mean piston speed per minute, feet	122.388
Mean steam pressure at boilers, pounds	93.4
Mean steam pressure at engine, pounds	91.9
Mean vacuum, inches	26.6
Mean pressure discharge main according to gauge in engine room, pounds	121.5
Mean pressure discharge main according to gauge in engine room, feet	280.7
Mean distance from center of gauge to level of water in pump well, feet	24.6
Allowance for friction in pumps (assumed), feet	2.31
Total head against pumps, feet	307.61
Mean temperature of water discharged, Fahr	61°
Total amount of feed water to boilers, pounds	116,245
Less 3½ % (assumed) for entrained water in steam, pounds	4,068
Actual amount of water evaporated, pounds	112,177
Total amount of coal consumed, pounds	16,473
Percentage moisture in coal	5
Total amount ashes and clinkers, pounds	1,664
Percentage ashes and clinkers in coal	10.1
Total amount of combustible, pounds	13,986
Mean barometer, inches	29.96
Mean temperature outside air, Fahr	78°

RESULTS—ENGINE.

CAPACITY.

During the trial of 24 hours the engine made 29,373 revolutions, which gives for the pumps an actual capacity under the working head, static and frictional, of 307.61 feet (no allowance for "slip"—or loss of action due to the supposed imperfect filling of the pump chambers—being made) of 182.212 × 29,373 = 5,352,113 gallons per 24 hours; or allowing 2 % for slip, 5,245,070 gallons per 24 hours. The contractors' guarantee was a delivery of 5,000,000 gallons of water per 24 hours, pump plunger displacement, against a head of 285 feet (this head understood to be static), at a piston speed of 120 feet per minute, or 20 revolutions per minute. The available static head at the time of the test was 278 feet only (lacking 7 feet or about 3 pounds in pressure of the specified head), the upper part of the reservoir not being entirely completed; but the steam pressure carried at the engine was only 92 pounds instead of 95 pounds as called for in the specifications and there is no doubt whatever of the ability of the engine to pump against the stipulated static head of 285 feet, or in fact a considerably greater one.

The capacity under the contractors' guarantee (no allowance being made in the contract for slip) based on plunger displacement is as follows:

At a piston speed of 120 feet per minute the engine would make 28,800 revolutions per 24 hours.

$$182.212 \times 28,800 = 5,247,705 \text{ gallons},$$

which is nearly 5 % in excess of the guarantee; or, allowing 2 % for slip, 5,142,751 gallons, which is nearly 3 % in excess of the guarantee.

DUTY.

Total number of gallons pumped during 24 hours	5,352,113
Weight of 1 gallon of water at 62° Fahr.	8.336
Total head in feet	307.61
Total weight of water weighed to boilers, pounds	116,245

$$\text{Duty} = \frac{5,352,113 \times 8.336 \times 307.61}{116,245}$$

= 118,061,733 foot-pounds per 1000 pounds feed-water, or 13,061,733 foot-pounds in excess of the guarantee, or nearly 12½ %.

Allowing 2 % for slip and 3½ % for entrained water in steam:

$$\text{Duty} = \frac{5,245,070 \times 8.336 \times 307.61}{112,177}$$

= 119,896,258 foot-pounds per 1000 pounds of steam, or 14,896,258 foot-pounds in excess of the guarantee, or about 14 %.

RESULTS—BOILERS.

Pounds of water evaporated per pound of coal from and at temperature of feed-water (mean 103.5° F.)........................ $\frac{116,245}{16,478}$ = 7.08 pounds.

Pounds of water evaporated per pound of coal from and at 212°........ 8.12 pounds.

Pounds of water evaporated per pound of combustible from and at temperature of feed-water .... $\frac{116,245}{13,986}$ = 8.31 pounds.

Pounds of water evaporated per pound of combustible from and at 212°.. 9.53 pounds.

A number of indicator cards from the high and low pressure cylinders were taken during the test, which showed an excellent distribution of the steam.

At the close of the test the engine was examined and none of the bearings were found to be unduly hot. The cross-head and link-bearings nearest the steam end were quite warm—this being due in a greater or less degree to their absorption of heat from the steam—but the main crank-shaft, crank-pin and beam-bearings were cool.

Appended hereto are the logs of the test.

Very respectfully,

(Signed) TRACY LYON,
Mechanical Engineer.

H. E. STEVENS,
Civil Engineer.

MAY [illegible] 192

[illegible] 1922